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IN MEMORIAM
George Davidson
1825-1911

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Professor of Geography
University of California

George Davis Jr

DUTY AND CAPACITY TESTS

OF

WORTHINGTON

HIGH DUTY

PUMPING ENGINES

ON

WATER WORKS AND PIPE LINE SERVICES

Apr 17/1903 ackg.
& sent sewage report.

HENRY R. WORTHINGTON
86 AND 88 LIBERTY STREET AND 145 BROADWAY
NEW YORK
1892

THE ENGINEERING PRESS,
277 PEARL ST., N. Y.

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NOTE.

THE history of the WORTHINGTON PUMPING ENGINE, dating from its invention in 1840 to its present state of development, has been frequently set before the public. In this brief review, containing a collection of duty and capacity tests, we merely desire further to emphasize the fact that not until the year 1884 was a "cut-off" ever used effectively in a direct-acting non-rotative engine.

In that year it was first used in connection with the "High-duty attachment," illustrated and described in many of the reports contained in this volume. This invention is one of the most novel developments of modern steam engineering. That its success during this brief period has been remarkable, needs no better proof than the subjoined list of places where the High Duty, as well as other types of Worthington Pumping Engines, have been furnished up to January 1, 1892.

A number of these tests have already appeared in pamphlet form, or in current engineering magazines in this country and England, but this is the first attempt to assemble them under one cover, and to put them before Engineers in more convenient form for reference.

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LIST OF PLACES WHERE

WORTHINGTON PUMPING ENGINES

HAVE BEEN FURNISHED FOR USE ON WATER WORKS SERVICE, THE YEAR WHEN THEY WERE ERECTED OR CONTRACTED FOR, AND THEIR DAILY CAPACITY IN U. S. GALLONS.

Location.	Capacity in U. S. Gallons
1860.	
Greenwood Cemetery, Brooklyn, N. Y.,	500,000
Harrisburg, Pa.,	2,000,000
1861.	
Mount Auburn, Mass.,	500,000
1863.	
Charlestown, Mass.,	5,000,000
1864.	
Wilmington, Del.,	500,000
1866.	
Annapolis, Md.,	500,000
1867.	
Charlestown, Mass.,	5,000,000
1868.	
Burlington, Vt.,	750,000
Burlington, Vt.,	750,000
Philadelphia, Pa. (Belmont Station),	5,000,000
Philadelphia, Pa. (Belmont Station),	5,000,000

Location.	Capacity in U. S. Gallons.
1869.	
Norristown, Pa.,	1,000,000
Prospect Park, Brooklyn, N. Y.,	1,000,000
Newark, N. J.,	5,000,000
Cambridge, Mass.,	5,000,000
1870.	
Newark, N. J.,	5,000,000
Philadelphia, Pa. (Delaware Station),	6,000,000
Philadelphia, Pa. (Roxborough Station),	5,000,000
Salem, Mass.,	5,000,000
Hudson River Hospital, Poughkeepsie, N. Y.,	1,000,000
Portland, Ore.,	1,000,000
1871.	
Columbia, Pa.,	1,500,000
Wayne, Pa.,	200,000
1872.	
State Hospital for Insane, Danville, Pa.,	1,500,000
Wilmington, Del.,	5,000,000
Charlestown, Mass.,	8,000,000
Providence, R. I.,	5,000,000
Philadelphia, Pa.,	2,000,000
Rahway, N. J.,	1,500,000
Rahway, N. J.,	1,500,000
Bowling Green, Ky.,	1,000,000
Zanesville, Ohio,	2,000,000
Poughkeepsie, N. Y.,	3,000,000
Marysville, Cal.,	750,000
1873.	
Philadelphia, Pa. (Belmont Station),	8,000,000
Jersey City, N. J. (Belleville Division),	8,000,000
Jersey City, N. J. (Belleville Division),	8,000,000
Conshohocken, Pa.,	1,000,000
Phœnixville, Pa.,	1,500,000
New Bedford, Mass..	3,000,000
Woburn, Mass.,	2,000,000
Salem, Mass.,	5,000,000
Waltham, Mass.,	1,500,000
Attleborough, Mass.,	1,000,000

Location.	Capacity in U. S. Gallons.
1874.	
Baltimore, Md.,	5,000,000
Baltimore, Md.,	5,000,000
Pasadena, Cal.,	750,000
Lindsay, Ont., Canada,	350,000
San Francisco, Cal.,	1,000,000
Newark, N. J.,	8,000,000
Cambridge, Mass.,	5,000,000
Baltimore, Md.,	3,000,000
Phœnixville, Pa., Iron Works,	2,500,000
Toledo, Ohio,	5,000,000
Toledo, Ohio,	5,000,000
Toronto, Canada,	5,000,000
Buffalo, N. Y.,	10,000,000
Montgomery, Ala.,	1,000,000
1875.	
Lincoln, Mass.,	450,000
Newark, N. J.,	3,000,000
Montreal, Canada,	11,000,000
Zanesville, Ohio,	3,000,000
Bristol, Pa.,	1,000,000
Brookline, Mass.,	2,000,000
Natick, Mass.,	1,500,000
College Point, N. Y.,	1,200,000
College Point, N. Y.,	1,300,000
Savannah, Ga.,	5,000,000
Fall River, Mass.,	5,000,000
Danville, Va.,	1,000,000
Staunton, Va.,	1,250,000
Bloomington, Ill.,	1,000,000
Racine, Wis.,	1,000,000
Michigan City, Ind.,	1,500,000
Yorkville, Canada,	1,500,000
Media, Pa.,	335,000
Cloverdale, Cal.,	500,000
1876.	
Willows, Cal.,	1,000,000
Los Angeles, Cal.,	1,000,000
Buffalo, N. Y.,	15,000,000
Toronto, Canada,	10,000,000

Location.	Capacity in U. S. Gallons.
Sandusky, Ohio,	3,000,000
Sandusky, Ohio,	2,000,000
Lowell, Mass.,	5,000,000
Jamaica Pond, Mass.,	2,000,000
Cleveland, Ohio,	10,000,000
Springfield, Ill.,	3,000,000
Danvers, Mass.,	2,000,000
Centennial Exhibition, Philadelphia, Pa.	5,000,000
Newton, Mass.,	5,000,000
Newton, Mass.,	1,000,000
Bordentown, N. J.,	500,000
1877.	
Colusa, Cal.,	1,000,000
Syracuse, N. Y.,	10,000,000
Boston, Mass.,	3,000,000
Mount Holly, N. J.,	500,000
Pittston, Pa.,	2,000,000
Kalamazoo, Mich.,	2,000,000
Bridgeton, N. J.,	1,000,000
Norwalk, O.,	2,500,000
Newcastle, Pa.,	3,000,000
San Antonio, Tex.,	1,500,000
Portland, Ore.,	3,000,000
Baltimore, Md.,	3,000,000
Rochelle, Ill.,	750,000
Willard Asylum, N. Y.,	1,500,000
Escondido, Cal.,	1,000,000
Sorel, Quebec, Canada,	1,000,000
1878.	
Elyria, Ohio,	1,500,000
Burlington, N. J.,	500,000
Paterson, N. J.,	3,000,000
Peru, Ind.,	2,000,000
Peru, Ind.,	1,500,000
Jacksonville, Ill.,	2,000,000
Lancaster, Pa.,	3,000,000
Haverhill, Mass.,	1,500,000
London, Ont., Canada,	2,000,000
Lewiston, Me.,	3,000,000
Lewiston, Me.,	3,000,000

Location.	Capacity in U. S. Gallons.
1879.	
Houston, Tex.,	3,000,000
New York, N. Y. (98th Street),	7,500,000
New York, N. Y. (98th Street),	7,500,000
Edgar Thompson Steel Co., Braddock, Pa.,	3,500,000
Edgar Thompson Steel Co., Braddock, Pa.,	3,500,000
Cambria Iron Works, Johnstown, Pa.,	3,500,000
Quincy, Ill.,	2,000,000
Plymouth, Mass.,	1,500,000
Jacksonville, Fla.,	3,000,000
Pennsylvania Steel Co., Steelton, Pa.,	6,000,000
New Brighton, Pa.,	750,000
Alameda, Cal.,	2,000,000
Jersey City, N. J.,	3,000,000
Youngstown, Ohio,	3,000,000
Brookline, Mass.,	1,000,000
Norfolk, Va.,	2,000,000
Norfolk, Va.,	2,000,000
Norfolk, Va.,	2,000,000
Union Stock Yard, Ill.,	2,000,000
Tewksbury, Mass.,	335,000
New Carlisle, Ind.,	335,000
1880.	
Woodstock, Ont., Canada,	1,000,000
Wheeling, W. Va.,	3,000,000
Cooperstown, N. Y.,	500,000
Cooperstown, N. Y.,	500,000
Macon, Ga.,	1,000,000
Alton, Ill.,	1,500,000
Brantford, Ont., Canada,	750,000
Nantucket, Mass.,	1,500,000
Rochelle, Ill.,	750,000
Jamaica Pond, Mass.,	750,000
Calumet and Hecla Mining Co., Michigan,	3,500,000
Edgar Thompson Steel Co., Braddock, Pa.,	3,500,000
Peoria, Ill.,	2,000,000
Peoria, Ill.,	2,000,000
St. Joseph, Mo.,	4,000,000
East Boston, Mass.,	750,000
Canton, Ohio,	1,000,000
Canton, Ohio,	2,000,000

LIST OF WORTHINGTON PUMPING ENGINES.

Location.	Capacity in U. S. Gallons.
Akron, Ohio,	1,500,000
Akron, Ohio,	1,000,000
Waltham, Mass.,	1,500,000
Cleveland, Ohio,	10,000,000
Newark, N. J.,	5,000,000
Yonkers, N. Y.,	3,500,000
Colorado Coal and Iron Co., Pueblo, Col.,	1,500,000
Philadelphia (Spring Garden Station),	10,000,000
Danvers, Mass.,	2,000,000
Albany & Rensselaer Iron and Steel Co., Troy, N. Y.,	3,000,000
Lancaster, Ohio,	1,500,000
Danville, Va.,	2,000,000
Buffalo, N. Y.,	15,000,000
Haverhill, Mass.,	1,500,000
Boston, Mass. (Sewerage),	25,000,000 +
Boston, Mass. (Sewerage),	25,000,000 +

1881.

Short Hills, N. J.,	200,000
Quincy, Ill.,	2,000,000
Annapolis, Md.,	1,250,000
Auburn, Me.,	750,000
Wilmington, N. C.,	2,000,000
Wellsville, Ohio,	1,500,000
St. Charles, Mo.,	1,500,000
Greenwood Cemetery, Brooklyn, N. Y.,	500,000
Lowell, Mass.,	750,000
Somerville, N. J.,	500,000
McKeesport, Pa.,	1,500,000
McKeesport, Pa.,	1,500,000
Portland, Ore.,	500,000
Richmond, Va.,	6,000,000
Chillicothe, Ohio,	1,500,000
Keokuk, Iowa,	1,500,000
Tombstone, Ariz.,	500,000
St. Louis, Mo.,	500,000
Litchfield, Ill.,	500,000
Otis Iron and Steel Co., Cleveland, Ohio.,	750,000
Muncie, Ind.,	1,500,000
Edgar Thompson Steel Co., Braddock, Pa.,	3,500,000

Location.	Capacity in U. S. Gallons.
Edgar Thompson Steel Co., Braddock, Pa.,	3,500,000
Edgar Thompson Steel Co., Braddock, Pa.,	3,500,000
Edgar Thompson Steel Co., Braddock, Pa.,	3,500,000
Winston, N. C.,	500,000
Standard Oil Co., Cleveland, Ohio.,	5,000,000
Standard Oil Co., Cleveland, Ohio.,	5,000,000
Peabody, Mass.,	2,500,000
Peabody, Mass.,	2,500,000
Savannah, Ga.,	7,500,000
Hackensack Water Co., Hackensack, N. J.,	3,000,000
Hackensack Water Co., Hackensack, N. J.,	3,000,000
Long Branch, N. J.,	2,500,000
Cambria Iron Works, Johnstown, Pa.,	5,000,000
St. Joseph, Mo.,	3,000,000
1882.	
Mt. Vernon, Ohio,	1,000,000
Mt. Vernon, Ohio,	1,000,000
Painesville, Ohio,	2,500,000
Morrison, Ill.,	2,000,000
State Hospital for Insane, Danville, Pa.,	1,500,000
New Haven, Conn.,	6,000,000
Burden Iron Works, Troy, N. Y.,	3,500,000
Winona, Minn.,	2,000,000
Winona, Minn.,	2,500,000
Atlantic City, N. J.,	1,500,000
Atlantic City, N. J.,	1,500,000
Pottstown, Pa.,	1,500,000
Springfield, Ill.,	4,000,000
Lowell, Mass.,	750,000
Cambridge, Mass.,	1,000,000
Hamilton, Ont., Canada,	1,250,000
Ishpeming, Mich.,	750,000
Gunnison, Col.,	1,500,000
Tyler, Tex.,	1,000,000
Locust Mountain, Pa.,	2,225,000
Locust Mountain, Pa.,	2,225,000
West Chester, Pa.,	1,000,000
Hot Springs, Ark.,	1,000,000
Pernambuco, Brazil,	1,500,000
Phœnixville, Pa.,	1,500,000

Location.	Capacity in U. S. Gallons.
New York City, N. Y. (High Bridge),	5,000,000
New Brunswick, N. J.,	5,500,000
San Diego, Cal.,	1,000,000
1883.	
Freeland, Pa.,	350,000
Sunbury, Pa.,	1,250,000
Sunbury, Pa.,	1,500,000
Tunkhannock, Pa.,	350,000
Westminster, Md.,	450,000
Painesville, Ohio	1,000,000
Negaunee, Mich.,	1,500,000
Mandan, Dak.,	500,000
State Penitentiary, Huntsville, Tex.,	350,000
St. Thomas, Ont., Canada,	350,000
Zumbrota, Minn.,	500,000
Standard Oil Co., Bayonne, N. J.,	5,000,000
Standard Oil Co., Bayonne, N. J.,	5,000,000
Bradford, Pa.,	750,000
Hackensack Water Co., Hackensack, N. J.,	2,000,000
Hackensack Water Co., Hackensack, N. J.,	2,000,000
Yonkers, N. Y.,	1,250,000
Portland, Ore.,	5,000,000
Portland, Ore.,	5,000,000
Phœnixville, Pa.,	500,000
Olean, N. Y.,	2,000,000
Stratford, Ont., Canada,	2,500,000
Milton, Pa.,	2,000,000
Winfield, Kan.,	1,000,000
Portsmouth, Va.,	6,000,000
Alliance, Ohio,	1,500,000
Elizabethtown, N. J.,	2,500,000
Cambria Iron Works, Johnstown, Pa.,	5,000,000
Galesburg, Ill.,	1,500,000
La Grange, Ill.,	1,500,000
Philadelphia, Pa. (Spring Garden Station),	15,000,000
Philadelphia, Pa. (Spring Garden Station),	15,000,000
Philadelphia, Pa. (Roxborough Station),	7,500,000
Lancaster, Pa.,	6,000,000
Fergus Falls, Minn.,	1,250,000
Beaver Falls, Pa.,	3,500,000

Location.	Capacity in U. S. Gallons.
Des Moines, Iowa,	6,000,000
New Bethlehem, Pa,	1,000,000
Nashville, Tenn.,	2,500,000
Nashville, Tenn.,	2,500,000
Nashville, Tenn.,	2,500,000
Nashville, Tenn.,	2,500,000
Olean, N. Y.,	1,000,000
Pittsburgh (South Side), Pa.,	4,000,000
Bloomsburg, Pa.,	1,000,000
Chamberlain, Dak.,	500,000
Cleburn, Tex.,	1,250,000
Terrell, Tex.,	1,000,000
Georgetown, Tex.,	750,000
Cleveland, Ohio,	10,000,000
Corsicana, Tex.,	750,000
Corsicana, Tex.,	750,000
Amesbury, Mass.,	2,000,000
Macon, Ga.,	750,000
Galveston, Tex.,	2,000,000
Sandwich, Ill.,	1,000,000
San Marcos, Tex.,	750,000
Brookville, Pa.,	1,000,000
Waterbury, Conn.,	1,000,000
Wilkesbarre, Pa.,	1,000,000
Newburgh, N. Y.,	1,500,000
Long Island City, N. Y.,	2,000,000

1884.

Temple, Tex.,	750,000
Canandaigua, N. Y.,	1,250,000
Nashua, N. H.,	3,000,000
Brooklyn, N. Y. (Springfield Station),	2,000,000
Sayre, Pa.,	1,500,000
Columbia, S. C.,	1,000,000
Princeton, N. J.,	500,000
Belton, Tex.,	1,000,000
Lampasas, Tex.,	500,000
Lampasas, Tex.,	500,000
Cortland, N. Y.,	1,500,000
Whitman, Mass.,	1,250,000

Location.	Capacity in U. S. Gallons.
Dunkirk, N. Y.,	1,500,000
Negaunee, Mich.,	1,500,000
Sauk Centre, Minn.,	1,250,000
Newton, Kan.,	3,000,000
Gonzales, Tex.,	450,000
Aylmer, Ont., Canada,	450,000
Woodstock, Ont., Canada,	1,000,000
Sheffield, Ala.,	200,000
Bridgeport, Conn.,	4,000,000
Woonsocket, R. I.,	2,000,000
Irvington, N. Y.,	500,000
Atlantic City, N. J.,	1,500,000
South Abington, Mass.,	1,250,000
Augusta, Ga.,	3,000,000
Bethlehem Iron Co., Pa.,	3,000,000
Jersey Shore, Pa.,	750,000
Amesbury, Mass.,	2,000,000
Wahpeton, Dak.,	1,000,000
Sioux Falls, Dak.,	1,500,000
Crystal, W. Co.,	1,500,000
Newark, N. J.,	8,000,000
Braddock, Pa.,	2,500,000
Braddock, Pa.,	2,500,000
Passaic, N. J.,	2,000,000
Trenton, N. J.,	6,000,000
West Haven, Conn.,	500,000
Canton, Ill.,	750,000
New Orleans Exposition,	1,500,000
New Orleans Exposition,	1,500,000
1885.	
*New Bedford, Mass.,	5,000,000
Manheim, Pa.,	500,000
Flatbush, N. Y.,	2,500,000
Red Bank, N. J.,	1,500,000
Newcastle, Pa.,	3,000,000
Bethlehem, Pa.,	500,000
Quincy, Ill.,	2,000,000
Houlton, Me.,	500,000

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* High Duty Engine.

Location.	Capacity in U. S. Gallons.
Houlton, Me.,	500,000
Corvallis, Ore.,	500,000
Muncie, Ind.,	1,000,000
Hegewisch, Ill.,	1,500,000
Lake View, Ill.,	5,000,000
Brooklyn, N. Y. (Watts' Pond),	3,000,000
Mamaroneck, N. Y.,	200,000
Jacksonville, Fla.,	2,500,000
Pierre, Dak.,	750,000
Cambria Iron Works, Johnstown, Pa.,	5,000,000
Belleville, Ill.,	1,250,000
Belleville, Ill.,	1,250,000
San Antonio, Tex.,	6,000,000
Fort Cœur de Alêne, Idaho,	750,000
Adams, N. Y.,	1,000,000
Gravesend, N. Y.,	2,000,000
Gravesend, N. Y.,	2,000,000
Bethlehem, Pa.,	750,000
Lansing, Mich.,	1,500,000
Lansing, Mich.,	1,500,000
San Francisco, Cal. (South Side),	1,250,000
Marshall, Mo.,	1,000.000
Marshall, Mo.,	1,000,000
Orange, N. J.,	1,750,000
Auburn, Me.,	1,000,000
Moberly, Mo.,	600,000
Moberly, Mo.,	750,000
Fulton, N. Y.,	1,500,000
Whitman, Mass.,	1,250,000
Homer, N. Y.,	1,000,000
Greenwood Cemetery, Brooklyn, N. Y.,	1,000,000
*Montreal, Canada,	12,000,000
Richfield Springs, N. Y.,	1,000,000
Riverside and Oswego Mills,	2,500,000
Stoughton, Mass.,	500,000
Waco, Tex.,	1,500,000
San Antonio, Tex.,	1,250,000
Putnam, Conn.,	1,000,000
Paola, Kan.,	750,000

*High Duty Engine.

Location.	Capacity in U. S. Gallons.
Paola, Kan.,	750,000
Oak Park, Ill.,	2,000,000
Crystal Plate Glass Co., Missouri,	2,500,000
Jamestown, N. Y.,	750,000
Waterford, N. Y.,	1,000,000
Norway, Me.,	650,000
Presque Isle, Me.,	650,000
Wenonah, N. J.,	500,000
Elyria, Ohio,	1,000,000
Neilsville, Wis.,	500,000
Astrakhan, Russia,	2,000,000
Aylesbury, England,	500,000
Mexboro, England,	400,000
Sevenoaks, England,	500,000
London, England (Kent Co.),	500,000

1886.

Location.	Capacity in U. S. Gallons.
Cape May City, N. J.,	1,000,000
Morristown, N. J.,	500,000
Freeland, Pa.,	500,000
Union Bridge, Md.,	450,000
Orangeburgh, S. C.,	450,000
Meridian, Miss.,	2,500,000
Eureka, Ill.,	350,000
Fort Worth, Tex.,	2,500,000
Guadalajara, Mexico,	1,000.000
Dallas, Tex.,	2,225,000
Dallas, Tex.,	2,225,000
Tuxedo Park, N. Y.,	750,000
Shelbyville, Ill.,	1,000,000
Shelbyville, Ill.,	1,000,000
Long Branch, N. J.,	3,000,000
Bethlehem Iron Co., Pa.,	3,000,000
Texarkana, Tex.,	1,000,000
Brunswick, Ga.,	1,000,000
Shelbyville, Ind.,	1,000,000
Shelbyville, Ind.,	1,000,000
Warsaw, Ind.,	1,000,000
Warsaw, Ind.,	1,000,000
Selma, Ala.,	1,000,000

Location.	Capacity in U. S. Gallons.
Selma, Ala.,	1,000,000
East Greenwich, R. I.,	750,000
East Greenwich, R. I.,	750,000
Caldwell, Kan.,	1,500,000
Marion, Iowa,	1,000,000
Marion, Iowa,	1,000,000
Bridgeport, Conn.,	4,000,000
Anthony, Kan.,	750,000
Anthony, Kan.,	750,000
Stockton, Cal.,	2,000,000
Freeland, N. J.,	1,000,000
Wooster, Ohio,	750,000
Crawfordsville, Ind.,	1,500,000
Crawfordsville, Ind.,	1,500,000
Crawfordsville, Ind.,	2,000,000
East St. Louis, Ill.,	3,000,000
East St. Louis, Ill.,	3,000,000
Waterloo, N. Y.,	1,000,000
Waterloo, N. Y.,	1,000,000
*Abington and Rockland, Mass.,	2,000,000
Abington and Rockland, Mass.,	1,250,000
Exeter, N. H.,	1,000,000
Exeter, N. H.,	1,000,000
Westerly, R. I.,	875,000
Westerly, R. I.,	875,000
Cincinnati, Ohio,	6,000,000
Lawrence, Kan.,	1,225,000
Lawrence, Kan.,	1,225,000
Houston, Tex.,	5,000,000
White Plains, N. Y.,	1,250,000
Lakewood, N. J.,	1,000,000
Pittsburgh, (South Side) Pa.,	1,500,000
Memphis, Tenn.,	4,000,000
Bethlehem Iron Co., Pa.,	3,000,000
Clarinda, Iowa,	1,125,000
Clarinda, Iowa,	1,125,000
Crystal Plate Glass Co., Missouri,	2,000,000
Crystal Plate Glass Co., Missouri,	2,000,000
Bath, Me.,	3,000,000

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* High Duty Engine.

Location.	Capacity in U. S. Gallons.
Wilmington, N. C.,	1,500,000
Easton, Pa.,	2,000,000
Conshohocken, Pa.,	1,000,000
Escanaba, Mich.,	1,250,000
Escanaba, Mich.,	1,250,000
Cornwall, Ont., Canada,	1,250,000
Cornwall, Ont., Canada,	1,250,000
Palatka, Fla.,	1,000,000
Palatka, Fla.,	1,000,000
*Hackensack Water Co., New Milford, N. J.,	5,000,000
Chillicothe, Ohio,	2,000,000
Brownwood, Tex.,	1,000,000
Brownwood, Tex.,	1,000,000
Kansas City, Mo.,	9,000,000
Macon, Ga.,	1,500,000
*Davenport, Iowa,	5,000,000
Trenton, N. J.,	2,000,000
Muncie, Ind.,	1,500,000
Sherman, Tex.,	750,000
Sherman, Tex.,	750,000
Cape May City, N. J.,	1,000,000
Morristown, N. J.,	500,000
Freeland, Pa.,	500,000
Union Bridge, Md.,	450,000
Orangeburgh, S. C.,	450,000
Meridian, Miss.,	2,500,000
Eureka, Ill.,	350,000
Astrakhan, Russia,	2,000,000
Havana, Cuba,	1,000,000
London, England (Southwark and Vauxhall Co.),	500,000
Tourcoing, France,	1,500,000
Tourcoing, France,	1,500,000
*London, England (West Surrey Station).	1,200,000
Mexboro, England,	400,000
Zamora, Spain,	350,000
*London, England (Chelsea Co.),	18,000,000
Berlin, Germany,	750,000
Berlin, Germany,	750,000
Southampton, England,	900,000

* High Duty Engine.

Location.	Capacity in U. S. Gallons.
*Bournemouth, England,	2,500,000
†Folkestone, England,	750,000
Swansea, South Wales,	900,000
1887.	
Tacony, Pa.,	1,500,000
Tioga, Pa.,	500,000
Monteagle, Tenn.,	150,000
Pittsburgh, Kan.,	1,000,000
San Buenaventura, Cal.,	200,000
Sea Cliff, N. Y.,	350,000
Industrial Reformatory, Huntington, Pa.,	1,000,000
Littleton, N. H.,	500,000
Paris, Tex.,	1,000,000
Paris, Tex.,	1,000,000
Canton, Ohio,	3,000,000
Bristol, Cal.,	2,000,000
Wilkesbarre, Pa.,	4,000,000
Belleville, Ont., Canada,	1,500,000
Belleville, Ont., Canada,	1,500,000
Rochester, Minn.,	1,000,000
Rochester, Minn.,	1,000,000
Ft. Smith, Ark.,	3,000,000
Salisbury, N. C.,	1,000,000
Salisbury, N. C.,	1,000,000
Newark, N. J.,	2,000,000
Ashtabula, Ohio,	1,500,000
Ashtabula, Ohio,	1,500,000
Nebraska City, Neb.,	1,000,000
Nebraska City, Neb.,	1,000,000
Chillicothe, Mo.,	1,000,000
Chillicothe, Mo.,	1,000,000
Montclair, N. J.,	750,000
Montclair, N. J.,	750,000
Greenbush, N. Y.,	1,000,000
Greenbush, N. Y.,	1,000,000
Memphis, Tenn.,	6,000,000
*Cedar Rapids, Iowa,	2,000,000
Crookston, Minn.,	1,250,000

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* High Duty Engine. † Triple Expansion Engine.

Location.	Capacity in U. S. Gallons.
Hurley, Wis.,	750,000
St. Louis, Mo.,	5,000,000
St. Louis, Mo.,	5,000,000
St. Louis, Mo.,	5,000,000
St. Louis, Mo.,	5,000,000
Duluth, Minn.,	3,000,000
Watsontown, Pa.,	1,000,000
*Calumet and Hecla Mining Co., Michigan,	10,000,000
Sheboygan, Wis.,	1,500,000
Sheboygan, Wis.,	1,500,000
Huntington, W. Va.,	2,500,000
Elgin, Ill.,	1,500,000
Elgin, Ill.,	1,500,000
Catonsville, Md.,	750,000
Portage City, Wis.,	1,000,000
Portage City, Wis.,	1,000,000
Cincinnati, Ohio,	25,000,000
East St. Louis, Ill.,	3,000,000
Cortland, N. Y.,	1,500,000
Millville, N. J.,	1,250,000
Jamestown, N. Y.,	1,000,000
Sanford, Fla.,	500,000
Bessemer, Mich.,	750,000
Kamloops, B. C.,	750,000
Grafton, Mass.,	500,000
Haddonfield, N. J.,	1,000,000
Frankfort, Ky.,	1,500,000
South Pueblo, Col.,	2,000,000
San Antonio, Tex.,	1,500,000
Union Stock Yard, Ill.,	2,000,000
Fort Scott, Kan.,	1,250,000
Fort Scott, Kan.,	1,250,000
Middletown, Del.,	500,000
Bridgewater, Mass.,	1,000,000
Bridgewater, Mass.,	1,000,000
Skowhegan, Me.,	1,000,000
Skowhegan, Me.,	1,000,000
Pittsburgh, Pa.,	2,000,000
Geneva, N. Y.,	1,500,000

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* High Duty Engine.

Location.	Capacity in U. S. Gallons.
Cincinnati, Ohio,	500,000
North Easton, Mass.,	750,000
North Easton. Mass.,	750,000
Marblehead, Mass.,	1,000,000
Fresno, Cal.,	2,000,000
Litchfield, Ill.,	1,000,000
Sanford, Fla.,	1,250,000
Winfield. Kan.,	1,250,000
White Plains, N. Y.,	500,000
Ocala, Fla.,	1,000,000
Sumter, S. C.,	1,000,000
Tampa, Fla.,	1,000,000
Tampa. Fla.,	1,000,000
*Minneapolis, Minn.,	15,000,000
*Minneapolis, Minn.,	15,000,000
*Norristown, Pa.,	5,000,000
Lake City, Fla.,	500,000
Orebro, Sweden,	1,000,000
Tenby, South Wales,	400,000
Leyden, Holland,	3,000,000
*Bradford, England,	2,000,000
*Bradford, England,	2,000,000
Birkenhead, England,	400,000
Blaydon Colliery, England,	2,700,000
*Northampton, England,	1,750,000
*Northampton, England,	1,750,000
Bristol Docks, England,	500,000
Bristol Docks, England,	500,000
*London, England (West Middlesex Co.),	26,000,000
London, England (Grand Junction Co.),	1.500,000
*London, England (New River Co.),	10,000,000
Island of Malta (Goya),	500,000
London, England (West Middlesex Co.),	350,000
Palermo, Sicily,	400,000
*Portsmouth, England,	5,000,000
Swansea, South Wales,	500,000
Lismore, Australia,	500,000
Lismore, Australia,	500,000
South America,	600,000

*High Duty Engine.

1888.

Location.	Capacity in U. S. Gallons.
*Atlantic City, N. J.,	5,000,000
Alameda, Cal.,	1,500,000
St. Johnland, N. Y.,	1,500,000
*Leavenworth, Kan.,	6,000,000
Sam Christian Gold. Hydraulic, Ltd.,	2,225,000
Sam Christian Gold. Hydraulic, Ltd.,	2,225,000
St. Clair Tunnel (G. T. Ry.),	5,000,000
*Akron, Ohio,	3,000,000
Texarkana, Tex.,	1,000,000
Hammond, Ind.,	750,000
Hammond, Ind.,	750,000
Beaver Dam, Wis.,	1,000,000
Beaver Dam, Wis.,	1,000,000
Green Island, N. Y.,	2,000,000
Green Island, N. Y.,	2,000,000
Oshkosh, Wis.,	1,500,000
Cedar Falls, Iowa.,	1,000,000
Pontiac, Mich.,	1,500,000
Pontiac, Mich.,	1,500,000
Yonkers, N. Y.,	2,000,000
Manitowoc, Wis.,	2,000,000
Manitowoc, Wis.,	2,000,000
Berlin, Ont., Canada,	1,000,000
Berlin, Ont., Canada,	1,000,000
Franklin, Pa.,	1,000,000
*Elmira, N. Y.,	6,000,000
Trenton, Mo.,	300,000
Everett, Mass.,	1,000,000
Dennison, Ohio,	1,000,000
Dennison, Ohio,	1,000,000
Winchester, Mass.,	300,000
Sandusky, Ohio,	7,500,000
Wellsville, N. Y.,	1,000,000
Watkins, N. Y.,	250,000
Tuscaloosa, Ala.,	1,000,000
Tuscaloosa, Ala.,	1,000,000
Manchester, Va.,	750,000
Manchester, Va.,	750,000
Sandy Hill, N. Y.,	750,000

* High Duty Engine.

Location.	Capacity in U. S. Gallons.
Sandy Hill, N. Y.,	750,000
Newark, N. J.,	5,000,000
Beloit, Kan.,	1,000,000
Hillsboro, Ill.,	750,000
Grand Haven, Mich.,	1,000,000
Managua, Nicaragua,	100,000
Baltimore, Md.,	5,000,000
Evansville, Ind.,	2,500,000
Tuxedo Park, N. Y.,	1,000,000
Binghamton Asylum, N. Y..	1,500,000
*Birmingham, Ala.,	5,000,000
*Memphis, Tenn.,	10,000,000
*Memphis, Tenn.,	10,000,000
*Memphis, Tenn..	10,000,000
Addison, N. Y.,	750,000
Biddeford, Me.,	4,500,000
Aberdeen, Dak. (Sewerage),	1,500,000
New York Dyewood Co., Brooklyn, N. Y.,	6,000,000
Bryn Mawr, Pa.,	3,000,000
*Brooklyn, N. Y.,	10,000,000
*Brooklyn, N. Y.,	10,000,000
Herrington, Kan.,	1,000,000
Stanley Freehold Gold Mines, Ltd.,	1,500,000
Asheville, N. C ,	750,000
Brooklyn, N. Y. (Baisley's),	6,000,000
Matthiesen & Wiecher's Sugar Refinery,	15,000,000
*Chicago, Ill. (Hyde Park),	12,000,000
East Portland, Ore.,	1,250,000
Wakefield and Narragansett Pier, R. I.,	2,500,000
Spring Grove Cemetery, Cincinnati, Ohio,	1,000,000
Morristown, N. J.,	350,000
Broken Bow, Neb.,	1,000,000
Inglewood, Cal.,	1,250,000
Wiarton, Ont., Canada,	750,000
*Parana, Argentine Republic,	1,125,000
*Parana, Argentine Republic,	1,125,000
*Parana, Argentine Republic,	1,125,000
*Parana, Argentine Republic,	1,125,000
*Odessa, Russia.	5,000,000

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* High Duty Engine.

Location.	Capacity in U. S. Gallons.
*Odessa, Russia,	5,000,000
Northampton, England,	400,000
Montmartre, France,	2,000,000
Bath, England,	300,000
L'Orient, France,	5,000,000
Liverpool, England,	1,200,000
Orenburg, Russia,	900,000
Orenburg, Russia,	900,000
Tenby, South Wales,	1,200,000
Clevedon, England,	600,000
London, England (Kent Co.),	600,000
St. Petersburg, Russia,	10,000,000
St. Petersburg, Russia,	10,000,000
St. Petersburg, Russia,	10,000,000
St. Petersburg, Russia,	10,000,000
St. Petersburg, Russia,	10,000,000
Managua, Nicaragua,	100,000
Bristol Docks, England,	1,000,000
London, England (Lambeth Co.),	6,000,000
*Sydney, New South Wales,	15,000,000
Tunbridge, England,	750,000
*Paris Exposition, 1889,	6,000,000
*Monte Video, South America,	5,000,000
Folkestone, England,	1,000,000

1889.

Location.	Capacity in U. S. Gallons.
Shelter Island, N. Y.,	300,000
*Chattanooga, Tenn.,	8,500,000
*Quincy Mining Co., Michigan,	8,000,000
Crozer Steel and Iron Works,	3,000,000
Lackawanna Iron and Coal Co.,	3,000,000
Tarboro, N. C.,	300,000
South Boston, Va.,	300,000
Taylor, Tex.,	1,000,000
Orangeburgh, S. C.,	1,250,000
American Wire Co., Cleveland,	2,000,000
Greenport, N. Y.,	1,000,000
Sag Harbor, N. Y.,	1,000,000
Cartersville, Ga.,	1,250,000
Cartersville, Ga.,	1,250,000

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* High Duty Engine.

Location.	Capacity in U. S. Gallons.
Sheffield, Ala.,	750,000
Park Ridge, Ill.,	200,000
Fernandina, Fla.,	1,250,000
Brantford, Ont., Canada,	2,000,000
Marshall, Mich.,	1,000,000
Marshall, Mich.,	1,000,000
Marblehead, Mass.,	1,000,000
Cobourg, Ont., Canada,	1,000,000
Cobourg, Ont., Canada,	1,000,000
Canandaigua, N. Y.,	1,250,000
Santa Barbara, Cal.,	1,000,000
Tarrytown, N. Y.,	1,250,000
Mount Holly, N. J.,	1,000,000
Baker City, Ore.,	750,000
Wytheville, Va.,	750,000
Mammoth Springs, Ark.,	700,000
Crete, Neb.,	1,000,000
*Solvay Process Co., Syracuse, N. Y.,	5,000,000
Reading, Pa.,	5,000,000
* †Port Perry, Pa.,	3,000,000
Nutley, N. J.,	500,000
Grafton, Mass.,	1,000,000
Pictou, Nova Scotia,	1,000,000
Washington Court House, Ohio,	1,500,000
Washington Court House, Ohio,	1,500,000
Mount Vernon, N. Y.,	2,000,000
Monroe, Mich.,	1,500,000
Monroe, Mich.,	1,500,000
Culiacan, Mexico,	450,000
Alexandria, Minn.,	1,000,000
*Nashville, Tenn.,	10,000,000
Fort Keogh, Mont.,	500,000
Somerville, Mass.,	2,000,000
Goldsboro, N. C.,	1,000,000
Goldsboro, N. C.,	1,000,000
*Jersey City, N. J.,	5,000,000
Marysville, Kan.,	750,000
Marysville, Kan.,	500,000
Wellington, Kan.,	1,000,000

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* High Duty Engine. † Triple Expansion Engine.

Location.	Capacity in U. S. Gallons.
New Dorp, Staten Island, N. Y.,	250,000
Bradford, Mass.,	1,000,000
Rouse's Point, N. Y.,	1,000,000
Laredo, Tex.,	1,500,000
Summit, N. J.,	1,000,000
Summit, N. J.,	1,000,000
Pittsburgh Plate Glass Works,	1,750,000
Mazatlan, Mexico,	1,500,000
Fort Worth, Tex.,	2,000,000
*Little Rock, Ark.,	5,000,000
Nebraska City, Neb.,	500,000
Richards Paper Co., Gardiner, Me.,	3,250,000
Ishpeming, Mich.,	3,000,000
Illinois Steel Co., Chicago, Ill.,	4,312,500
Illinois Steel Co., Chicago, II.,	4,312,500
Illinois Steel Co., Chicago, Ill.,	4,312,500
Illinois Steel Co., Chicago, Ill.,	4,312,500
Illinois Steel Co., Chicago, Ill.,	4,312,500
Illinois Steel Co., Chicago, Ill.,	4,312,500
Illinois Steel Co., Chicago, Ill.,	4,312,500
Illinois Steel Co., Chicago, Ill.,	4,312,500
†Kettering, England,	1,000,000
*Oxford, England,	4,250,000
†Lisbon, Portugal,	3,000,000
†Calcutta, India,	6,500,000
†Calcutta, India,	6,500,000
†Barry & Cadoxton, Wales,	500,000
†Barry & Cadoxton, Wales,	500,000
South Africa,	750,000
South Africa,	750,000
Windsor, England,	500,000
†Bombay, India,	10,000,000
†Bombay, India,	10,000,000
*Cia Huanchaca de Bolivia,	6,000,000
London, England (Richmond Drainage),	1,500,000
London, England (Richmond Drainage),	1,500,000
London, England (Richmond Drainage),	1,500,000
London, England (Richmond Drainage),	1,500,000
Goya, Island of Malta,	150,000

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* High Duty Engine. † Triple Expansion Engine.

Location.	Capacity in U. S. Gallons.
Alexandria, Egypt,	1,200,000
London, England (West Middlesex Co.),	7,500,000
Jamu, India,	700,000
*Bournemouth, England,	2,000,000
Hemel Hemstead, England,	900,000
Nagpur, India,	2,000,000
Orenburg, Russia,	900,000
Swansea, South Wales,	900,000
Utrecht, Holland,	500,000
Newbabelsberg, Germany,	200,000
Cape Town, South Africa,	600,000
South Africa,	600,000
South Africa,	600,000
South Africa,	600,000
South Africa,	600,000
South Africa,	600,000
South Africa,	600,000
Island of Malta,	125,000
Tiel, Holland,	500,000
*Berlin, Germany,	5,000,000
*Berlin, Germany,	5,000,000
†Rotterdam City, Holland (City Sewerage),	30,000,000
1890.	
Florence, Ala.,	1,500,000
Florence, Ala.,	1,500,000
Tacoma, Wash.,	1,000,000
Tacoma, Wash.,	1,000,000
Brooklyn, N. Y.,	5,000,000
Woodland, Cal.,	1,125,000
Seattle, Wash.,	1,500,000
Seattle, Wash.,	1,500,000
Monmouth Park, N. J.,	500,000
*New York, N. Y. (98th Street),	10,000,000
Portland, Ore. (City Park),	350,000
Vicksburg, Miss.,	1,000,000
Danville, Pa.,	3,000,000
St. Lawrence State Hospital, Ogdensburg, N. Y.,	1,600,000
Miles City, Mont.,	1,000,000
*Baltimore, Md.,	5,000,000

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* High Duty Engine. † Triple Expansion Engine.

Location.	Capacity in U. S. Gallons.
Shreveport, La.,	2,000,000
Short Hills, N. J.,	1,000,000
Albina, Ore.,	750,000
Albina, Ore.,	750,000
Brunswick, Ga.,	2,000,000
Olympia, Wash.,	1,000,000
Olympia, Wash.,	1,000,000
Vallejo, Cal.,	1,000,000
Oldtown, Me.,	1,500,000
Massillon, Ohio,	1,000,000
Veazie, Me.,	1,500,000
Sioux Falls, Dak.,	3,000,000
St. Thomas, Ont., Canada,	2,000,000
St. Thomas, Ont., Canada,	2,000,000
Bessemer, Mich.,	1,000,000
Kansas City, Mo.,	1,500,000
Tallapoosa, Ga.,	750,000
Tallapoosa, Ga.,	750,000
Brooklyn, N. Y.,	1,500,000
Cincinnati, Ohio,	2,000,000
Brooklyn, N. Y.,	1,000,000
Crozer Iron Co.,	3,000,000
Puyallup, Wash.,	500,000
Mt. Vernon, N. Y.,	1,000,000
Sewickley, Pa.,	350,000
Richmond, Ky.,	1,000,000
Richmond, Ky.,	1,000,000
San Antonio, Tex.,	1,500,000
San Antonio, Tex.,	1,125,000
Kansas City, Kan.,	7.500,000
Albany, N. Y.,	1,000,000
American Iron Works, Pittsburgh, Pa.,	5,000,000
Fort Madison, Iowa,	1.250,000
Cooperstown, N. Y.,	1,000,000
Willard State Hospital, N. Y.,	2,000,000
*Peoria, Ill.,	7,000,000
*Peoria, Ill.,	7,000,000
*Peoria, Ill.,	7,000,000
Brockton, Mass.,	3,000,000

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* High Duty Engine.

Location.	Capacity in U. S. Gallons.
*Norfolk, Va.,	10,000,000
*Norfolk, Va.,	5,000,000
Portsmouth, Ohio,	3,000,000
Monongahela Water Co., Pittsburgh, Pa.,	10,000,000
*St. Louis, Mo.,	20,000,000
*St. Louis, Mo.,	20,000,000
*Hackensack Water Co., N. J., (Weehawken),	4,000,000
Langhorne, Pa.,	1,000,000
Mt. Tom Sulphite Pulp Co., Mass.,	4,000,000
Wellman Iron and Steel Co., Thurlow, Pa.,	4,000,000
Marquette, Mich.,	3,000,000
East Portland, Ore.,	3,000,000
Stratford, Ont.,	2,000,000
Elmhurst, Ill.,	350,000
Plainfield, N. J.,	2,000,000
Plainfield, N. J.,	3,000,000
Gravesend, N. Y.,	2,000,000
Johannesburg, South Africa,	750,000
Johannesburg, South Africa,	750,000
†Hong Kong, China,	60,000
†Poona, India,	3,000,000
†Rosario, South America,	5,000,000
†St. Petersburg, Russia,	4,500,000
Johannesburg, South Africa,	250,000
†Newport, England,	1,000,000
Ilkeston, England,	1,000,000
*†London, England (City Sewerage),	24,000,000
†Cirencester, England,	150,000
*†London, England (Lambeth),	12,000,000
†London, England,	1,200,000
†Mexico City, Mexico,	5,000,000
†Mexico City, Mexico,	5,000,000
†Mexico City, Mexico,	5,000,000
*Sydney, New South Wales,	6,000,000
†Delhi, India,	3,000,000
†Delhi, India,	3,000,000
†Shrewsbury, England,	2,500,000
Den Helder, Holland,	600,000
Hennbont, France,	400,000

* High Duty Engine. † Triple Expansion Engine.

Location.	Capacity in U. S. Gallons.
Farrington, England,	200,000
Dacca, India,	2,000,000
Lizy-Sur-Ourcq, France,	500,000
1891.	
Paterson, N. J.,	6,000,000
Holmesburg, Pa ,	1,000,000
Lakewood, N. J.,	750,000
Nyack, N. Y.,	750,000
Steubenville, Ohio,	4,000,000
Mamaroneck, N. Y.,	750,000
*Shamokin, Pa.,	1,400,000
*Lowell, Mass.,	12,000,000
Thornton, Ill.,	250,000
Eagle Pass, Tex.,	1,500,000
Newton, Pa.,	500,000
*Quincy Mining Co., Mich.,	10,000,000
Tyler, Tex.,	650,000
Dawson, Ga.,	750,000
Dawson, Ga.,	300,000
Floral Park, L. I.,	100,000
Bellefonte, Pa.,	1,500,000
*Hackensack Water Co., New Milford, N. J.,	10,000,000
Hinsdale, Ill.,	750,000
Milford, Mass.,	2,250,000
Independence, Ore.,	500,000
Brookline, Mass.,	1,500,000
Quincy, Ill.,	4,000,000
Quincy, Ill.,	6,000,000
Uvalde, Tex.,	500,000
Santa Ana, Cal.,	1,000,000
Santa Ana, Cal.,	1,000,000
East Orange, N. J.,	3,000,000
Ebensburg, Pa.,	500,000
Woodbury, N. J.,	1,500,000
Pottsville, Pa.,	650,000
Columbia, S. C.,	1,500,000
Columbia, S. C.,	1,500,000
Buchanan, Va.,	500,000

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* High Duty Engine.

Location.	Capacity in U. S. Gallons.
Scranton, Pa.,	2,000,000
Aberdeen, Wash.,	1,125,000
Mount Vernon, N. Y.,	1,000,000
Hancock, N. Y.,	500,000
Coshocton, Ohio,	1,125,000
Galesburg, Ill.,	2,000,000
Amesbury, Mass.,	350,000
Americus, Ga.,	1,500,000
†World's Columbian Exposition, Chicago, Ill.,	5,000,000
*World's Columbian Exposition, Chicago, Ill.,	12,500,000
World's Columbian Exposition, Chicago, Ill,	15,000,000
World's Columbian Exposition, Chicago, Ill.,	7,500,000
Pierre, S. D.,	1,000,000
Geneva, N. Y.,	1,500,000
*Toledo, Ohio,	15,000,000
Tarrytown, N. Y.,	1,000,000
Madera, Cal.,	750,000
Palouse, Wash.,	600,000
Long Island City, N. Y.,	2,000,000
*Nashville, Tenn.,	10,000,000
Newburgh, N. Y.,	1,500,000
Newburgh, N. Y.,	1,500,000
La Grange, Tex.,	500,000
Petersburg, Ill.,	750,000
Sanford, Me,	350,000
Danbury, Conn.,	1,300,000
La Porte, Ind.,	2,500,000
New Rochelle, N. Y.,	750,000
Sault Ste. Marie, Mich.,	2,500,000
New Rochelle, N. Y.,	1,250,000
New Rochelle, N. Y.,	2,000,000
Far Rockaway, N. Y.,	2,000,000
Bridgeport, Conn.,	10,000,000
Douai, France,	2,000,000
Canterbury, England,	1,500,000
Newcastle, Australia,	750,000
Newcastle, Australia,	750,000
†Reading, England,	2,400,000
North Seaton, England,	1,300,000

* High Duty Engine. † Triple Fxpansion Engine.

Location.	Capacity in U. S. Gallons.
†Rotterdam, Holland,	3,750,000
†Rotterdam, Holland,	3,750,000
†Folkestone, England,	1,575,000
†Folkestone, England,	1,575,000
Grimsby, England,	1,000,000
†St. Albans, England,	850,000
†London, England (West Middlesex Co.),	30,000,000
Sydney, New South Wales,	500,000
†St. Petersburg, Russia,	4,000,000
*†London, England (City Sewerage),	22,000,000
Suffolk County Asylum, England,	500,000
Suffolk County Asylum, England,	500,000
†Galatz, Roumania,	500,000
†Hubli, India,	155,000
†Hubli, India,	155,000
†London, England (West Middlesex Co.),	900,000
†London, England (West Middlesex Co.),	900,000
Paramatta, New South Wales,	430,000
Paramatta, New South Wales,	430,000
†Bombay, India (City Sewerage),	18,000,000
†Bombay, India (City Sewerage),	18,000,000
†Bombay, India (City Sewerage),	18,000,000
†Bombay, India (City Sewerage),	18,000,000
†Gibraltar, Gibraltar,	1,200,000
†Kettering, England,	870,000
†London, England (Grand Junction Co.),	6,000,000
†Berkhampstead, England,	690,000
1892.	
Clifton Forge, Va,	500,000
Pasadena, Cal.,	1,250,000
Hammond, Ind.,	3,000,000
Beardstown, Ill.,	750,000
Beardstown, Ill.,	750,000
Highland Park, Ill.,	250,000
Cambridge, N. Y.,	500,000
Alliance, O.,	2,000,000
Rouse's Point, N. Y.,	1,000,000
Whittier, Cal.,	750,000
Bluefield, W. Va.	1,000,000

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* High Duty Engine. † Triple Expansion Engine.

Location.	Capacity in U. S. Gallons.
†Concord, N. H,	2,000,000
Hudson, Wis.,	1,300,000
Royersford, Pa.,	800,000
Royersford, Pa.,	800,000
Vineyard Haven, Mass.,	1,500,000
Lapeer, Mich.,	100,000
West New Brighton, N. Y.,	350,000
Birmingham, Conn.,	1,500,000
*Philadelphia, Pa. (Spring Garden),	20,000,000
*Lancaster, Pa.,	10,000,000
Colfax, Wash.,	1,000,000
Brooklyn, N. Y. (East New York),	450,000
Aitkin, Minn.,	500,000
Charlevoix, Mich.,	500,000
Charlevoix, Mich.,	500,000
Long Beach, Cal.,	1,000.000
Kendallville, Ind.,	1,000,000
Hudson River State Hospital, Poughkeepsie, N. Y.,	650,000
Maquoketa, Iowa,	1,250,000
Watkins, N. Y.,	650,000
Watkins, N. Y.,	650,000
Salem, Mass.,	3,000,000
Salem, Mass.,	3,000,000
Eugene, Ore.,	1,500,000
Whitestone, N. Y.,	1,000,000
*Marlboro, Mass.,	2,500,000
†Asheville, N. C.,	1,250,000
Asheville, N. C.,	750,000
Hudson, N. H.,	250,000
Northport, N. Y,	250,000
Northport, N. Y.,	250,000
Cadillac, Mich.,	1,500,000
Cadillac, Mich.,	1,500,000
Schenectady Locomotive Works, N. Y.,	1,250,000
Albina, Ore.,	2,000,000
*Cleveland, O.,	15,000,000
*Erie, Pa.,	12,000,000
Glasgow, Ky.,	200,000
*Houston, Tex.,	8,000,000

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* High Duty Engine. † Triple Expansion Engine.

Location.	Capacity in U. S. Gallons.
Ithaca, Mich.,	500,000
Ithaca, Mich.,	500,000
Caldwell, Tex.,	500,000
Baker City, Ore.,	500,000
Huntington, N. Y.,	400,000
Huntington, N. Y.,	400,000
Everett, Pa.,	500,000
Haddonfield, N. J.,	450,000
Monroe, La.,	1,250,000
Monroe, La.,	1,250,000
Pullman, Wash.,	500,000
Champaign, Ill.,	1,000,000
Bloomfield, Iowa,	500,000
Everett, Wash.,	1,000,000
Macon, Ga.,	2,000,000
*Shamokin, Pa.,	1,500,000
Gilroy, Cal.,	500,000
Chester, Pa.,	1,250,000
Chester, Pa.,	1,250,000
Brownsville, Pa.,	1,000,000
Brooklyn, N. Y. (Ridgewood),	5,000,000
*Monongahela Water Co., Pittsburgh, Pa.,	5,000,000
†Villaseca, Spain (Irrigation),	7,000,000
Hautmont, France,	275,000
†Raipur, India,	960,000
†Raipur, India,	960,000
†Reading, England,	720,000
†Reading, England,	720,000
*†London, England (New River Co.),	5,400.000
†Kishineff, Russia,	480,000
†Kishineff, Russia,	290,000
†Powell Duffryn Steam Coal Co.,	1,000,000
†Warminster, England,	400,000
†Brünn, Austria,	1,500,000
†Zamora, Spain,	400,000
Mymensingh, India,	360,000
Mymensingh, India,	360,000
*†London, England (West Surrey),	2,600,000
†Simla, India,	550,000

* High Duty Engine. † Triple Expansion Engine.

Location.	Capacity in U. S. Gallons
†Simla, India,	550,000
†Madura, India,	600,000
† Madura, India,	600,000
† Madura, India,	600,000
† Burdwan, India,	1,100,000
† Burdwan, India,	1,100,000
† Rotterdam, Holland,	18,000,000
† Trichinopoly, India,	1,250,000
† Trichinopoly, India,	1,250,000
† Trichinopoly, India,	1,250,000
† Trichinopoly, India,	520,000
† Trichinopoly, India,	520,000
† Lucknow, India,	3,000,000
† Lucknow, India,	3,000,000
† Cawnpore, India,	3,000,000
† Cawnpore, India,	3,000,000
† Lucknow, India,	3,000,000
† Lucknow, India,	3,000,000
† Cawnpore, India,	3,000,000
† Cawnpore, India,	3,000,000
† Tanjore, India,	850,000
† Tanjore, India,	850,000
† Tanjore, India,	850,000
† Sholapur, India,	1,500,000

To January 1, 1893, the total contract pumping capacity of Worthington Pumping Engines, the location of which is shown in the foregoing list, is two billion, nine hundred and twenty-three million, ninety thousand U. S. gallons in twenty-four hours.

In addition to the above, seven Worthington High Duty Oil Line Pumping Engines, having an aggregate daily capacity of 9,000,000 gallons, against a head of 2,300 feet, have been built for The National Transit Co.'s Pipe Lines.

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† Triple Expansion Engine.

HYDRAULIC WORKS, BROOKLYN, N. Y.

TEST ENGINE.

3,000,000 GALLONS CAPACITY.

"EXPERIMENTS ON A DIRECT-ACTING STEAM PUMP,"*

BY JOHN GEORGE MAIR, M. INST. C. E.

In the autumn of 1885, the author casually heard that a system of pumping, invented by Mr. C. C. Worthington, of the firm of Henry R. Worthington, of New York, was in use in the United States, enabling a Worthington direct-acting steam-pump to work with as high a rate of expansion as any type of crank and fly-wheel engine, and at the same time exert a steady and uniform pressure on the pump-plunger. He therefore determined to investigate and test its working. The motions of both a steam-piston and a water-plunger being rectilinear, a connecting-rod, crank and fly-wheel having a rotative motion, are superfluous except for the purposes of expansive working or controlling the length of stroke. Mr. E. D. Leavitt, Jr., who has a large and varied practice as a hydraulic engineer in America, explained to the author generally the peculiarity of the design of the engine, expressed himself in the highest terms of its mechanical efficiency, and kindly offered to assist in any experiments it was proposed to carry out.

The author took as an assistant, Mr. Henry Smith, Assoc. M. Inst. C. E., and in order that no question should be raised as to the accuracy of the necessary testing instruments, a circular orifice, through which to measure the air-pump discharge, three Kew-tested thermometers, an indicator, and also three tested Bourdon gauges for water and steam pressures, were sent from England.

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*Proceedings of the Institution of Civil Engineers, Vol. LXXXVI., Paper No. 2,187.

The inventor kindly placed an engine and its boiler entirely at the service of the author, and expressed a wish that the trials should be as complete and exhaustive as it was possible to make them. The engine was at work at Brooklyn, N. Y., and was put up solely for experimental purposes. It pumped out of a well, and through weighted relief valves back to the well, so that trials could be made which would have been impossible had the engine been performing the ordinary duty at a water-works. To pump about 1,700 gallons a minute through weighted and spring valves is a more difficult service than pumping against a head of water in a main. It was, therefore, evident that whatever results were obtained on the trials, they could be readily repeated and improved upon in practice. Nearly twenty-five years have passed since the first Worthington compound-condensing engine was erected and set to work in America; since then great improvements have been made, and now these machines pump 40 per cent. of the total water supply of the United States. The system, however, is not much known in England, and so little attention has it attracted, that there are no records of it in the Proceedings of this Institution, or in those of the Institution of Mechanical Engineers. In fact, it has not even been alluded to by the authors of the various papers on pumping engines that have been published from time to time. Practically the system consists of two independent engines and pumps lying side by side, the motion of one engine actuating the valves of the other. The delivery of water from the pumps is almost absolutely uniform, and although an air vessel is usually placed on the discharge chamber, it is generally water-logged, and the author could not tell the difference in working either with or without air.

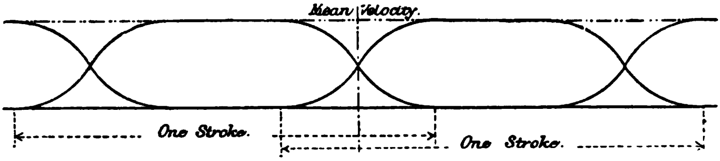

FIG. 1. FLOW FROM WORTHINGTON PUMP.

Figure 1 represents, approximately, the flow from a Worthington pump at each point of the stroke. As soon as one pump begins to slow down at the end of the stroke the other pump starts, so that by combining the flow it will be seen how uniform it is. With any pump driven by a crank and connecting rod, and even when two pumps are coupled on one crank shaft at right angles, great variation exists in the quantity of water delivered at different parts of the stroke, owing to the varying speed of the pistons, necessitating an air vessel being placed on the delivery main.

The delivery from a compound rotative engine, with cranks at right angles, working two double-acting pumps, supposing the connecting rod to be indefinitely long, is shown by Fig. 2. The deliveries are added together and shown in full lines; the variation of flow in this case is sufficient to make the pressures fluctuate to such an extent that accidents are very liable to occur when working without air. The Author, in his own practice, has met with many cases where accidents have happened to the pump-work and rising mains, when, through carelessness, no air was in the vessel; but with the uniform delivery of the type of twin-pumps before described an air-vessel is not needed, and it is this uniform delivery that permits the use of the engine for pumping through the oil-pipe lines where the friction in the mains amounts to 3,450 feet head at normal speed. With the single or double-acting pumps first used for this service, where the flow ceased at the end of the stroke, the pressure gauge fluctuated hundreds of pounds on the square inch with a corresponding result of broken pipes and pumps.

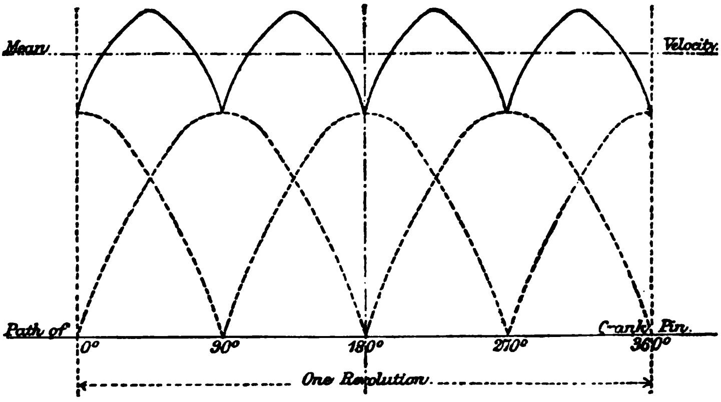

FIG. 2. VELOCITY DIAGRAMS.
TWO DOUBLE-ACTING PUMPS WITH CRANKS AT RIGHT ANGLES.

The oil-pipe lines are of different diameters and lengths, and, taking as an example one that came under the personal notice of the author, namely, 6 inches in diameter and about 30 miles long, through which two 10-inch double-plunger pumps were forcing oil, the main would contain, if filled with oil at a specific gravity of 0.87, over 750 tons, and as this weight may be considered as attached to the pump-piston, a very simple calculation will show what excessive pressures are set up when such a weight is moved at a variable velocity, and also as the pressure in the pump is nearly all due to friction in the main, which increases or decreases practically as the square of the speed of the flow in it, it can be seen that the only system of pumping capable of working with safety is that in which the delivery from the pump is uniform and regular at every part of the stroke. There are now on the oil lines some sixty or seventy compound condensing engines of various powers up to 600 or 800 H. P. The service is a peculiar one, and the difficulties that have been overcome reflect the greatest credit on the engineers of the line.

The Worthington engine just referred to, although as economical in fuel as an ordinary Cornish engine, and more so if the first cost and the expense of foundations and houses is taken into account, can be beaten in economy of fuel by a well-designed compound rotative-engine working at a high rate of expansion. Mr. C. C. Worthington therefore applied himself to attach to his engine a form of compensation which would absorb or store up the excess of power at the steam end during the first part of the stroke, and give it out again during the last part of the stroke, when owing to expansion the steam-pressure falls below the water-pressure.

Now the main point to be observed in designing such an arrangement, is to obtain a perfectly uniform pressure on the pump plunger, so as to get a steady delivery of water. To effect this, compensators of many varied forms were schemed, and an experimental engine was made that would work up to about 150 H. P., and a boiler arranged especially to supply it with steam. As it was almost impossible to obtain from the water-works sufficient water for the engine, a well was sunk, the entire plant with experiments having cost about £10,000. The engine was worked for about a year and a half continuously, and found to be such a perfect success that several are now at work, and many others are being made on the system that was in practice found best.

If the steam-pressure diagrams of an expansive compound engine are combined together, it will be found that there is an excess of pressure *a b* at the commencement of the stroke (Plate 8, Figs. 1) over the mean pressure decreasing to half-stroke, and after that point there is an increasing deficiency of pressure *b c*. This variation with a rotative engine is taken up by the fly-wheel, but in the high-duty Worthington engine there are two small cylinders (by preference oscillating) which are attached to the piston-rod, containing water or air under pressure. Referring to Plate 8, Figs. 2, it will be readily seen that the excess of work *a b*, which is a maximum at the commencement of the stroke and decreases to nothing at half-stroke, is taken up by these small cylinders. Directly after half-stroke, when the steam pressure is below the water pressure, they give out work *h k*, which increases to the end of the stroke, so that if the work absorbed or given out in the compensators is combined with the steam diagrams, a perfectly steady pressure-line is obtained, and the engine makes its stroke at a uniform speed, so that a straight pump-diagram is obtained. The diagrams, Plate 8, Figs. 1, were taken from a high-duty pumping engine, working under ordinary service at New Bedford, Mass., U. S. A., the steam being expanded during the time it was taken some 10 or 12 times.

Engine Trials.—These trials were all carried out in a similar manner to those before made by the author.* Plate 8, Figs. 3, give the general arrangement, plan of the boiler, engine, and pump, together with the position and details of the measuring tanks. The engine and pump are shown in Plate 8, Fig. 4. The feed water was measured in a cast-iron pipe, Plate 8, Fig. 5, with an overflow pipe in it, and its contents to the level of the pipe were weighed on tested scales many times over, the temperature being noted each time, so that the quantity of water in the pipe which was used as a feed measuring tank may be relied on as accurate. From the pipe the water was run into a wooden tank, out of which it was taken by the feed donkey and pumped into the boiler. Mr. C. C. Worthington placed one of his water-meters between the feed-pump and the boiler, and the meter readings agreed within ¼ per cent. with the measurements made by the author.

The boiler was of the Corliss type, vertical, 5 feet 4 inches in diameter by 14 feet high, with vertical tubes; and as the heat went direct from the fire through the tubes, and so heated the steam above the water-level, the steam was slightly superheated. A thermometer was fixed in the steam-pipe in the engine house, the readings of which are given in the tables. The steam-pipe went across a yard in the open air, but being well covered with non-conducting composition, and the steam being slightly superheated, condensation to any marked extent was prevented. The steam-jackets drained into a tank, which was carefully measured, and when full the condensed water was discharged into a drain, and the time noted. The working steam, after leaving the engine, passed through the eduction pipe to an independent air-pump and condenser, worked by a separate engine. Both the feed-donkey and the air-pump engine were supplied with steam from a separate boiler, so that, in taking the efficiency of the engine into account, the work done by these pumps should be deducted. Their having a separate steam-supply, did not, of course, affect the heat used by the main engine itself, but only the efficiency, that is, the relation of the indicated horse-power to the pump horse-power. The steam from the main engine, after being condensed and passing through the air-pump was delivered through a short length of pipe to the discharge-tank (Plate 8, Fig. 6), where it was gauged through a circular orifice 3 inches in diameter. The temperatures of injection and air-pump discharge were read, and the head measured every quarter of an hour. Eight new indicators, made by the American Steam Gauge Company (and which were checked with the English one) were on the steam cylinders fixed close up to each head, and the diagrams were averaged by ordinates in New York, and checked by planimeters in England. Two counters were on the engine, which checked each other, and two tested water-pressure gauges were fixed on the delivery main.

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* Minutes of Proceedings Inst. C. E., Vols. LXX. and LXXIX.

Five assistants were in the engine-room, and four in the boiler-house. A ship's chronometer was used for the time, and every quarter of an hour throughout all the test gongs were sounded, one in the engine-room and one in the boiler-house, so that all observations were taken at the same instant, and the author took personal observations all round every half hour, so that no error could have crept in. Such detailed care was, however, not necessary, as the rejected heat was measured, and that gives the best check on the boiler-supply. The stroke was kept the full length, touching the cylinder-heads each time; and so regularly did the engine run that, for each trial, all observations were almost exact counterparts of each other. Independently of measuring the heat-supply, many interesting experiments were made; the engine was slowed down until it made one double stroke in a minute and a half. The pump had its pressure suddenly released, to show the safety of the engine, and the air-vessel was filled with air, and was also water-logged; the compensators were put out of gear; in fact, every experiment was tried that was of value. The author made nine full trials, and Mr. Smith made three more after the author had left New York. These trials were so regular that it is sufficient to give the details of three.

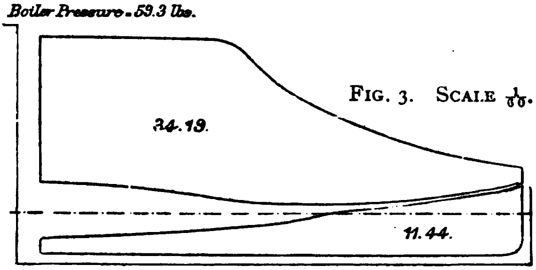

FIG. 3. SCALE $\frac{1}{60}$.

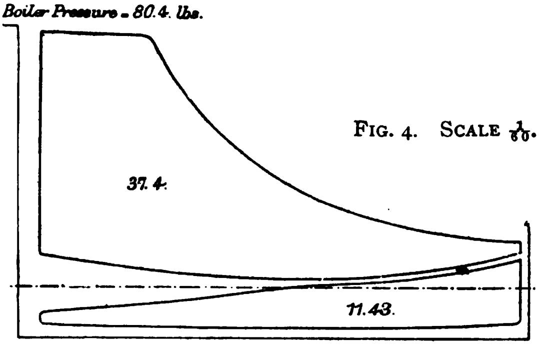

FIG. 4. SCALE $\frac{1}{60}$.

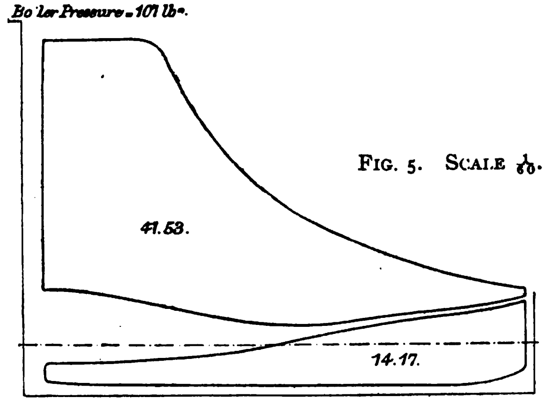

FIG. 5. SCALE $\frac{1}{60}$.

The absolute quantity of water delivered by the pumps could not be exactly ascertained; but even if the full displacement of the plunger was not made, it would not affect the results of the trials, as the pump horse-power was taken from the actual pressure in the delivery main (as recorded by the gauges tested in England) against the area of the plunger, all connections and by-passes being carefully shut off and plugged before the trials. At the end of each stroke a pause is made, which allows the pump-valves to close before the return stroke, and so prevents slip through them.

The average efficiency on the three trials is 91.5 per cent., but, from this has to be deducted the power it would require to work the air and feed-pumps, and taking this at 3½ per cent. would give a net result of 88 per cent. efficiency, or a higher value than is generally obtained by a crank and fly-wheel engine when the pump-valves are tight. This is what would be expected, as the pistons of the compensating cylinders and trunnions certainly produce less friction than the crank-shaft bearings, crank and cross-head pins, guide bars, eccentric straps, etc., of a fly-wheel engine.

The piston speed, as compared with the English practice, is very low, and naturally the repairs and renewals with these engines are of a most trivial character, even over long and extended periods of working. The foundations are simple, as the stresses are self-contained; in fact, the engine experimented with by the author was hardly on any foundation, and when doing 165 indicated horse-power, as it did on one of the trials, it was perfectly steady, and worked without noise or vibration.

The following is a summary of three trials:—No. 1 on December 24th, No. 2 on December 19th, and No. 3 on December 22d, 1885. (Figs. 3, 4, 5.)

No. of Trial,	1	2	3
Double strokes per minute,	45.0	39.26	40.10
Boiler-pressure, lbs.	59.3	80.4	101.0
Feed-water per minute (tank measurement) (Plate 8, Fig. 5), lbs.	34.12	30.33	36.26
Jacket drains per minute, "	4.22	4.15	4.57
Temperature of steam,	359°	376°	390°
Pressure on pump, including suction, lbs.	78.5	80.5	97.0
" in compensators, "	162.5	195.0	250.5
Mean pressure in high-pressure cylinder, lbs.	34.19	37.40	41.53
Mean pressure in low-pressure cylinder, lbs.	11.44	11.43	14.17
Temperature of injection,	57.18°	57.10°	57.30°
" air-pump discharge,	84.95°	81.06°	89.50°
Head over centers of orifice, ft.	1.727	1.802	1.397
Air-pump discharge per minute, lbs.	1,174.0	1,197.0	1,056.0
Injection water, "	1,144.0	1,171.0	1,024.0

No. of Trial, - - - - - - -	1	2	3
Heat passing through Engine per minute—			
T U from boiler, saturated steam through cylinders, - - - -	35,132.0	30,919.0	37,553.0
T U from boiler, superheat in steam,	853.0	772.0	906.0
" " condensation in jackets,	3,794.0	3,677.0	4,003.0
Total, - - - - - - -	39,779.0	35,368.0	42,462.0
Heat retained in condensed steam, -	1,585.0	1,283.0	1,822.0
" absorbed by injection water, -	31,769.0	28,057.0	32,972.0
" " indicated work, - -	5,096.0	4,621.0	5,579.0
" " radiation, - - -	440.0	440.0	440.0
Error - - - - - - - -	889.0	967.0	1,649.0
Total, - - - - - - -	39,779.0	35,368.0	42,462.0
Percentage of error to total heat passing through engine per minute,	2.2	2.7	3.8
Indicated H. P., - - - - - -	119.2	108.1	130.5
Pump H. P., - - - - - - -	109.3	97.9	120.4
Efficiency per cent., - - - -	91.7	90.6	92.3
Feed per I. H. P. per hour through cylinders, - - - - - -	15.05	14.53	14.57
Feed per I. H. P. per hour through jackets, - - - - - - -	2.12	2.30	2.10
Piston speed per minute per engine, ft.,	97.5	85.0	86.9
Boiler-pressure, - - - - - lbs.,	59.3	80.4	101.0
Number of expansions, - - - -	9.2	13.2	14.1
T U per I. H. P. per minute, - - -	334.0	327.0	325.0
Donkin's co-efficient, - - - -	273.5	265.2	260.6
T U per I.H.P. per minute calculated from the temperature of the air-pump discharge, - - - - - - - -	320.0	315.0	311.0
Lbs. of coal per I.H.P. per hour, supposing feed taken from hot well and the coal to give up 11,000 T U per lb.,* -	1.74	1.72	1.70
Duty in 1,000,000 foot-lbs. of water raised per 112 lbs. of coal taking 88 per cent. efficiency, - - - -	112.1	113.4	114.8
Disposal of Heat used—			
As indicated work, - - per cent.,	13.3	13.5	13.7
Rejected heat and error, - "	85.5	85.2	85.2
Radiation, - - - - - "	1.2	1.3	1.1

In order to ascertain exactly the dimensions of the engine and pump under test, the cylinder and pump-covers were taken off, and gauges made of the diameters of the four cylinders and their piston rods, and of the two pump plungers and their rods; these gauges were brought to London and measured with a standard Whitworth rule, the mean areas and lengths being as follows:

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* Minutes of Proceedings, Inst., C. E., Vol. LXX., p. 336.

Low-pressure cylinders, area,	1,013.0 sq. ins.
High “ “ “	251.0 “
Pump plungers, “ “	235.75 “
Stroke, length,	26.00 ins.
Clearance in low-pressure cylinder,	596.0 cu. ins.
“ high “ “	336.0 “

As before stated, the coal was not weighed; and in the table above 11,000 T U is taken, so that these trials can be compared with those previously made by the author.*

The engine worked perfectly on all the trials; was easily handled, and fully justified the opinion of its merits expressed by Mr. E. D. Leavitt, Jr., and the inventor is to be congratulated on having achieved a result which could only have been arrived at by a thorough knowledge of mechanics coupled with great perseverance and enterprise.

In conclusion, the author begs to tender his best thanks to Mr. C. C. Worthington; to his partner, Mr. W. A. Perry, and also to Mr. Barr, Mr. Root, and other members of the staff, for their kind assistance, and for the careful manner in which they carried out the instructions of the author relative to preparing the engine testing.

The Paper is accompanied by several diagrams, from which Plate 8, and the Figs. in the text have been prepared.

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*Minutes of Proceedings, Vols. LXX. and LXXIX.

J

NEW BEDFORD, MASS.

5,000,000 GALLONS CAPACITY.

ONE ENGINE.

REPORT OF TWO DUTY TESTS OF A WORTHINGTON HIGH DUTY PUMPING ENGINE AT THE NEW BEDFORD WATER-WORKS, IN JUNE, 1886.

NEW BEDFORD, MASS., November 11th, 1886.

To the New Bedford Water Board:

GENTLEMEN—The undersigned were appointed by the Water Board as a committee to subject the engine to the tests mentioned in the contract, and to see whether all the requirements thereof had been fulfilled.

The engine was built by Henry R. Worthington, and is essentially the same as the well known type so extensively used throughout the country, with the addition of an attachment which allows the steam to be used expansively, and by which greatly increased economy in the running of the engine has been secured.

As described by the builders, "the attachment consists briefly of two small oscillating cylinders attached to an extension of the plunger rod of the engine beyond the water end. These cylinders and their connecting pipes are filled with water. Compressed air from a storage tank is admitted at a suitable pressure to maintain a constant load upon the pistons in the cylinders through the medium of the interposed water.

" These pistons act in such a way with respect to the motion of the engine as to resist its advance at the commencement of the stroke and assist it at the end, the air, meanwhile, exerting its unvarying pressure at each point of the stroke.

" The two cylinders act in concert, and, being placed directly opposite each other, relieve the cross-head to which they are attached of any sliding frictional resistance, and the engine of any lateral strain.

"By thus alternately taking up and exerting power through the difference in the angle at which their force is applied with respect to the line of motion of the plunger rod, these two cylinders, in effect, perform the functions of a fly-wheel, but with the important mechanical difference that they utilize the constant pressure of compressed air instead of the energy of momentum. Their action is readily controlled, and their power can not only be exactly proportioned to the work to be overcome, but is entirely unaffected by the speed of the engine. The same amount of expansion can be obtained in the same engine, whether running at a piston speed of 10 feet per minute or at 150. This latter feature is of great importance, affecting, as it does, so favorably the economy of the engine when applied on any service where the demand is irregular or intermittent.

"Where such service is performed by a fly-wheel engine it is a well-known fact that the best economic results are attained only when the engine is running at its full rated capacity, and that its economy rapidly falls as its speed is decreased. With every change in the rate of rotation of the fly-wheel a corresponding change in the point of cut-off must be made.

"When the speed is decreased the steam must be made to follow further in the stroke of the pistons, thus reducing the expansion, and consequently the efficiency of the engine.

"The cut-off valves consist of semi-rotating plug-valves, placed in the admission ports of the cylinders, and operated by means of direct connections. Their action is secured without the use of any eccentrics, gears, or cams. When the point of the cut-off has been once fixed it need never be altered."

The engine worked under the disadvantage of having to take the steam through an unusually long steam pipe, 194 feet 3 inches in length, and the steam, which at the boiler was superheated, lost about 30 degrees of heat, so that at the engine it was 2 or 3 degrees below the temperature corresponding to the pressure in the boilers.

The general dimensions of the engine are as follows:

Diameter of high pressure cylinder, - - - -	18 in.
Diameter of low pressure cylinder, - - - - - -	36 in.
Diameter of plungers, - - - - - - - -	24 in.
Stroke, - - - - - - - - - - - -	26 in.

The vacuum is obtained by an independent condenser with a steam cylinder 6 inches diameter and 10 inches stroke, and a plunger 8½ inches diameter.

The cylinders are steam-jacketed and all water of condensation is returned to the boilers by a small self-regulating steam pump.

The diameter of the suction pipe is 24 inches, and the diameter of the discharge pipe is also 24 inches, connecting outside of the engine house with a 30-inch force main 1,879 feet long, which conveys the water to the distributing reservoir.

The dimensions of the boilers are as follows:

Diameter,	64 in.
Height,	13 ft. 7 in.
128 tubes 2 inches diameter,	10 ft. long.
Total heating surface,	1,190.1 sq. ft.
Grate surface of both boilers,	37.12 sq. ft.

The contract required that the engine should be capable of delivering 5,000,000 U. S. gallons in 24 hours, with a piston speed of not over 90 feet per minute.

The second requirement was that the engine should be capable of performing a duty of 100,000,000 based upon the plunger displacement and the amount of combustible consumed, the trial to continue twelve hours.

The third requirement was that the engine should perform a duty of 80,000,000 during a practical test covering six days of ten hours each, the engine to be run at a speed which shall insure the delivery of 5,000,000 gallons in 24 hours and all the fuel used to be taken into account.

According to the contract this duty test was calculated by the formula

$$\frac{P \times V \times H \times 100}{F}$$

wherein P represents the pounds of water delivered per stroke, as ascertained by calculation of plunger displacement, V represents the number of strokes made during the trial, H the head pressure as ascertained by a gauge placed on the force main or force chamber of the pump. To the indications of this gauge when running was added the measured distance from the centre of said gauge to the average surface of water in the pump well. No allowance was made for friction through foot valve and supply pipe, or any other part of the pumps, neither for any friction or increased resistance caused by bends or angles anywhere between said gauge and surface of water in pump well. F represents the number of pounds of coal actually consumed during the trial.

The engine was tested with regard to the first two requirements at the first trial, on June 1, 1886, and the requirements of the contract were satisfactorily fulfilled.

The trial was conducted in the following manner:

The fires were started at 7.45 A. M., and the engine was started at 8.50 A. M. The regular test was begun at 12 o'clock, noon, when the condition of the fires was carefully examined, and the test was continued for 12 hours and 5 minutes, when the condition of the fires was, upon careful examination, considered to be the same as at the beginning.

The boilers used were two upright Corliss boilers, the dimensions of which are given on the preceding page, and they were fired alternately at intervals of about 30 minutes for each boiler.

Observations were taken every 10 minutes of the reading of the counter and the pressure of the steam at the boiler, and also at the engine, and observations were taken every 30 minutes of the vacuum in the condenser, water pressure gauge, height of water in the pump well, pressure of air in the air chamber, height of water in boilers, reading of the meter which registered the quantity of water pumped into the boilers, the reading of the air pump counter, and the temperature of the feed water at its entrance to, and exit from the heater, the temperatures of the air pump delivery, the pump well, engine room, outside air, the steam at the boiler and at the engine, and the temperature of the gases before entering and after leaving the heater.

The water pressure gauge was tested by comparison with a standard gauge of the Crosby Steam Gauge Co., and was found to be accurate within the limits registered during the trial.

The height of water in the well was obtained by two large float gauges, tested before each trial, and also by actual measurement with a stick.

The temperature of the steam was taken by two thermometers inserted in the main steam pipe, one near the boilers and the other near the engine.

The latter was broken on the fourth day of the second test, so that these readings had to be discontinued.

The meter which registered the quantity of water delivered to the boilers was a ¾-inch Worthington meter, which we were informed had been tested under approximately the same pressure as that to which it was subjected during the trial, but subsequent tests have convinced us that the meter readings are not sufficiently reliable, and it is intended to have at some early date a careful test to ascertain the evaporate power of the boilers.*

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* The tests above referred to were made February 16th and 17th, 1887, with the following results:

Water actually evaporated per pound of dry coal from actual pressure and temperature, - - - - - - -	8.48 lbs.
Equivalent water evaporated per pound of dry coal from and at 212° Fahr.:	
Including feed water heater, - - - - -	9.63 lbs.
Excluding feed water heater, - - - - -	9 25 "

The barometer used was an aneroid placed in the engine room. It had been tested at the Royal Observatory at Greenwich.

Indicator diagrams from the steam cylinders were taken frequently. In order to ascertain the actual quantity of water pumped and the "slip" of the pumps, observations were taken at a weir carefully constructed at a point where the water enters the distributing reservoir. The length of the weir was 4 feet, and observations were taken every minute for periods of one and two hours at a time.

The results of this first test showed that the engine had pumped at a rate of 6,233,511 gallons in 24 hours at an average piston speed of 94 feet per minute, an excess of 19 per cent. above the first contract requirement, and that the engine had performed a duty of 102,108,759,* an excess of 2 per cent. above the second contract requirement.

The second test, to ascertain the duty during a period of six days, was begun on Monday, June 7th, 1886, at 8.55 A. M., and ended on Saturday, June 12th, 1886, at 6.44 P. M. The total running time was 61 hours 6 minutes, or an average of 10 hours 11 minutes per day. The shortest time was 10 hours, 4 minutes on the 12th, and the longest 10 hours, 20 minutes on the 8th.

Observations were taken every 30 minutes of the counter, steam pressure at boiler and at engine, vacuum, water pressure, height of water in pump well, height of water in boiler, readings of meter and of air pump counter. All the temperatures were observed once an hour.

The results of the second test showed that the engine had performed a duty during the six days of 79,238,160,† which is within less than one per cent. of the third contract requirement.

Thus, the first requirement was exceeded by 19 per cent., the second requirement was exceeded by 2 per cent., and in the third requirement there was a deficiency of less than 1 per cent.

In justice to the engine it should be stated that during the trial it was not convenient to subject the engine to a head of 132 feet *exclusive of friction*, as required by the contract, but as in the opinion of the committee the duty of the engine would have been greater under this increased head, they felt justified in advising the acceptance of the engine.

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* Evaporation of boilers during this test per pound of coal from temperature of injection to steam at temperature due pressure, - 7.38 lbs.

† Evaporation of boilers during this test per pound of coal from temperature of injection to temperature due pressure, - - - 6.74 lbs.

The "slip" of the pumps, or the difference between the plunger displacement and the actual quantity of water pumped, was found by the weir measurements to vary from 1.42 to 1.94 per cent., which indicates a very satisfactory performance.

The detailed results of the observations will be found in a concise form in the tables annexed to this report.

Respectfully submitted,

(Signed) WILLIAM ROTCH, R. C. P. COGGESHALL, WILLIAM R. BILLINGS, } *Committee.*

RECORD OF TWO DUTY TESTS OF ENGINE NO. 3 (WORTHINGTON), AT THE NEW BEDFORD WATER-WORKS.

	FIRST TEST.	SECOND TEST.
1. Date of trials, - - -	June 1, 1886.	June 7-12, 1886
2. Time of beginning trial, -	12.00 M.	8.55 A. M.
3. Time of ending trial, - -	12.05 A. M.	6.44 P. M.
4. Duration of trial, - -	12 h. 5 m.	61 h. 6 m.
5. Total counts ["revolutions"]	15,751	66,789
6. Total strokes, - - -	63,004	267,156
7. Average strokes per minute,	86.9	72.9
8. Total travel of plunger corresponding to full stroke, -	1,638,104 in.	6,946,056 in.
9. Total observed shortage in travel of plunger, - -	18.7 in.	795.4 in.
10. Displacement of plunger for one full stroke, - -	49.813275 gals.	49.813275 gals.
11. Average displacement per stroke during trials, - -	49.812706 gals.	49.807571 gals.
12. Distance from gauge to water in pump well, - - -	17.56 ft.	20.95 ft.
13. Head above gauge, - -	110.22 ft.	109.85 ft.
14. Total lift, - - - - -	127.78 ft.	130.80 ft.
15. Gallons pumped, - -	3,138,399	13,306,391
16. Rate per 24 hours, gallons, -	6,233,511	5,226,732
17. Contract requirement in 24 hours, gallons, - - -	5,000,000	5,000,000
18. Excess above contract requirement, - - -	19 per ct.	4 per ct

		FIRST TEST.	SECOND TEST.
19.	Temperature of water, - -	65.12 deg.	67.55 deg.
20.	Weight of water per cubic foot, - - - - -	62.34 lbs.	62.33 lbs.
21.	Weight of water per gallon,	8.3337 lbs.	8.3323 lbs.
22.	Weight of combustible burned, - - - -	3,273 lbs.	
23.	Weight of coal burned - -		18,302 lbs.

24. Duty of engine, $\frac{P\times V\times H\times 100}{F}$

$$\text{First trial,} = \frac{\overbrace{49.812706\times 8.3337}^{P}\times \overset{V}{63{,}004}\times \overset{H}{127.78}\times 100}{\underset{F}{3{,}273}} = 102{,}108{,}759$$

$$\text{Second trial,} = \frac{\overbrace{49.807571\times 8.3323}^{P}\times \overset{V}{267{,}156}\times \overset{H}{130.80}\times 100}{\underset{F}{18{,}302}} = 79{,}238{,}160$$

25.	Average pressure of steam in boilers,	114.69 lbs.	114.51 lbs.
26.	Average pressure of steam in pipe near engine, - - - -	114.36 lbs.	114.01 lbs.
27.	Average vacuum, - - - -	28.39 in.	28.46 in.
28.	Average atmospheric pressure by barometer, - - - -	30.03 in.	29.86 in.
29.	Average temperature of air in engine room, - - - - - -	76.24 deg.	83.73 deg.
30.	Average temperature of external air,	58.77 deg.	65.87 deg.
31.	Average temperature of feed water at entrance to heater, - - -	179.04 deg.	169.02 deg.
32.	Average temperature of feed water at exit from heater, - - -	231.08 deg.	231.31 deg.
33.	Average temperature of air pump delivery, - - - - -	83.60 deg.	85.06 deg.
34.	Average temperature of steam at boiler. - - - - -	370.96 deg.	374.04 deg.
35.	Average temperature of steam at engine, - - - -	346.28 deg.	341.28 deg.
36.	Average temperature of gases in flue before heater, - - -	361.40 deg.	360.37 deg.
37.	Average temperature of gases in flue after heater, - - - - -	259.80 deg.	261.58 deg.

		FIRST TEST.	SECOND TEST.
38.	Slip of pumps as tested by weir measurement, - - - -	Not observed.	1.70 per ct.
39.	Indicated horse-power, mean of four cards, - - - - - -	153.70	
40.	Indicated horse-power, mean of two cards, - - - - - -		134.02
41.	Horse-power in water pumped, pump measurement, no allowance for "slip," - - - - - -	135.15	120.02
42.	Work done by pumps in per cent. of indicated H. P., - - - -	87.93	89.55
43.	Combustible burned per hour per indicated H. P., - - - -	1.76 lbs.	
44.	Coal burned per hour per indicated H. P., - - - - - -		2.24 lbs.

BOILER TEST.

REPORT OF A TEST OF TWO UPRIGHT CORLISS BOILERS, AT THE PUMPING STATION OF THE NEW BEDFORD WATER-WORKS, TO DETERMINE THEIR EVAPORATIVE EFFICIENCY.

New Bedford, Mass., March 16th, 1887.

To the New Bedford Water Board:

Gentlemen—In accordance with instructions received from your Board, the undersigned respectfully present the following report of a test of the two upright Corliss boilers now used in connection with the new Worthington pumping engine.

On the morning of February 16th, 1887, the two boilers were supplying steam to the Worthington engine, which was then pumping the city's supply. At 8.30 A. M. the engine was stopped, the fires under the boilers were drawn, and the ash pits cleaned. At 9.12 A. M. new fires were started under each boiler.

At 9.30 A. M., when the test began, the steam pressure was 98 pounds at the boiler, as shown by the gauge. At 9.54 A. M. the engine was started, and ran continuously until 10.15 A. M., February 17th, 1887, the boiler pressure at that time being 111.5 pounds.

The height of the water in the boilers was carefully noted at the beginning and end of the trial, and the proper correction was made for the difference, which was very slight.

The coal used was Lehigh coal, known as "Lindermann's Sugar Loaf."

At the end of the trial all combustible coal was picked from the refuse and weighed. This weight (being less than 2 per cent.) was deducted from the gross amount of coal charged.

The gauges and thermometers had all been previously tested. The barometer was an Aneroid, which had been carefully tested and found correct.

The quantity of water fed to the boilers was measured in the following manner: A tank was placed upon a tested platform scale and supplied by the feed pump with water from the hot-well, the temperature of the water being noted at the time of weighing. The water, after being weighed, was allowed to flow into a lower tank, from which it was pumped by a second feed pump into the boilers.

The water was also measured by a Worthington meter, the same one which had been used in the previous trial of the engine.

Observations were taken every half hour of the steam pressure at the boilers, the height of water in the boilers, and the reading of the meter, and the quantity of water fed to the boilers was computed every thirty minutes.

Observations were taken every hour of the atmospheric pressure near the boilers, and the following temperatures: Steam at boilers, feed water, before and after passing through heater, flue gases, before and after passing through heater, air in boiler room, and external air.

Appended to this report we present the results tabulated in the form recommended by the Committee of the American Society of Mechanical Engineers.

Respectfully submitted.

WM. ROTCH,
WM. R. BILLINGS,
R. C. P. COGGESHALL.

RECORD OF A TEST OF TWO UPRIGHT CORLISS BOILERS, AT THE PUMPING STATION OF THE NEW BEDFORD WATER-WORKS.

Reported in form recommended by the Committee of the American Society of Mechanical Engineers.

1.	Date of trial,	February 16 and 17, 1887.
1a.	Time of beginning trial,	9.30 A. M.
2.	Duration of trial,	24 hours 45 minutes.

DIMENSIONS AND PROPORTIONS.

	Type of boiler,	Corliss vertical, internal fire-box.
	Number in use,	Two.
	Diameter of shell,	64 in.
	Length of shell,	13 ft. 7 in.
	Number of tubes, each boiler,	128
	Diameter of tubes,	1¾″ inside, 2″ outside.
	Length of tubes, exposed to water 6′ 3″, exposed to steam 3′ 9″,	10 ft.
	Kind of grate bars,	Bannister rocking grate.
	Total area of air space in grates, two boilers,	16.7 sq. ft.
	Area of chimney flue, 3 feet square,	9 sq. ft.
	Area of horizontal flue, connecting with chimney,	7.11 sq. ft.
	Height of chimney,	70 ft.
	Length of horizontal flue, connecting with chimney,	86 ft.
3.	Grate surface, circular, diameter 5′ 11⅞″ one boiler, 18.56 sq. ft. both boilers,	37.12 sq. ft.
3a.	Ratio, area of grate to area air spaces,	2.2 to 1
3b.	Area of least draught,	5.2 sq. ft.
4.	Water heating surface, both boilers,	749.84 sq. ft.
5.	Superheating surface, both boilers,	440.26 sq. ft.
5a.	Total heating surface, both boilers,	1,190.10 sq. ft.
5b.	Heating surface in feed water heater,	183.26 sq. ft.
6.	Ratio of water heating surface to grate surface,	20.2 to 1
6a.	Ratio of superheating surface to grate surface,	11.9 to 1
6b.	Ratio of total heating surface to grate surface,	32.1 to 1
6c.	Ratio of water heating surface to superheating surface,	1.7 to 1

AVERAGE PRESSURES.

7.	Steam pressure in boiler by gauge.	114.98 lbs.
8.	Absolute steam pressure,	129.53 lbs.
9.	Atmospheric pressure by barometer,	29.64 in.

AVERAGE TEMPERATURES.

11.	Of external air,	36.96° Fahr.
12.	Of boiler room,	81.12° Fahr.

13.	Of steam,	369.04° Fahr.
14.	Of escaping gases before passing feed water heater,	366.04° Fahr.
14a.	Of escaping gases after passing feed water heater,	262.64° Fahr.
15.	Of feed water before passing heater,	123.27° Fahr.
15a.	Of feed water after passing heater,	165.48° Fahr.

FUEL.

16.	Total amount of coal consumed,	7,422 lbs.
17.	Moisture in coal, 143 pounds,	$1\frac{92}{100}$ per ct.
18.	Dry coal consumed,	7,279 lbs.
19.	Total refuse dry, 708 pounds,	$9\frac{73}{100}$ per ct.
20.	Total combustible (weight of coal, item 18, less refuse, item 19),	6,571 lbs.
21.	Dry coal consumed per hour,	294 lbs.
22.	Combustible consumed per hour,	266 lbs.

WATER.

26.	Total weight of water pumped into boiler and apparently evaporated,	61,720.4 lbs.
26a.	Check on above by meter measurement,	57,768 lbs.
26b.	Error of meter,	—$6\frac{84}{100}$ per ct.
28.	Equivalent, water evaporated into dry steam from and at 212° Fahr.:	
	Including feed water heater,	70,071 lbs.
	Excluding feed water heater,	67,357 lbs.
29.	Equivalent, total heat derived from fuel in British Thermal units:	
	Including feed water heater,	67,667,777
	Excluding feed water heater,	65,047,129
30.	Equivalent water evaporated into dry stream from and at 212° Fahr. per hour:	
	Including feed water heater,	2,831 lbs.
	Excluding feed water heater,	2,721 "

ECONOMIC EVAPORATION.

31.	Water actually evaporated per pound of dry coal from actual pressure and temperature,	8.48 lbs.
32.	Equivalent, water evaporated per pound of dry coal from and at 212° Fahr.:	
	Including feed water heater,	9.63 lbs.
	Excluding feed water heater,	9.25 "

33. Equivalent, water evaporated per pound of combustible from and at 212° Fahr.:

Including feed water heater, - - 10.66 lbs.
Excluding feed water heater, - - 10.25 "

COMMERCIAL EVAPORATION.

34. Equivalent, water evaporated per pound of dry coal, with one-sixth refuse, at 70 pounds gauge pressure, from temperature of 100° Fahr. = item 33 multiplied by 0.7249:

Including feed water heater, - - 7.73 lbs.
Excluding feed water heater, - - 7.43 "

RATE OF COMBUSTION.

35. Dry coal actually burned per square foot of grate surface per hour, - - - - - - - - - 7.92 lbs.

36.	Consumption of dry coal per hour, coal assumed with one-sixth refuse.	per sq. ft. of grate surface, -	8.58 lbs.
37.		per sq. ft. of water heating surface,	0.425 lbs.
37a.		per sq. ft. of superheating surface,	0.724 lbs.
37b.		per sq. ft. of total heating surface,	0.268 lbs.
38.		per sq. ft. of least area for draught,	61.26 lbs.

RATE OF EVAPORATION.

39. Water evaporated from and at 212° Fahr. per sq. ft. of heating surface per hour, excluding feed water heater, 3.63 lbs.

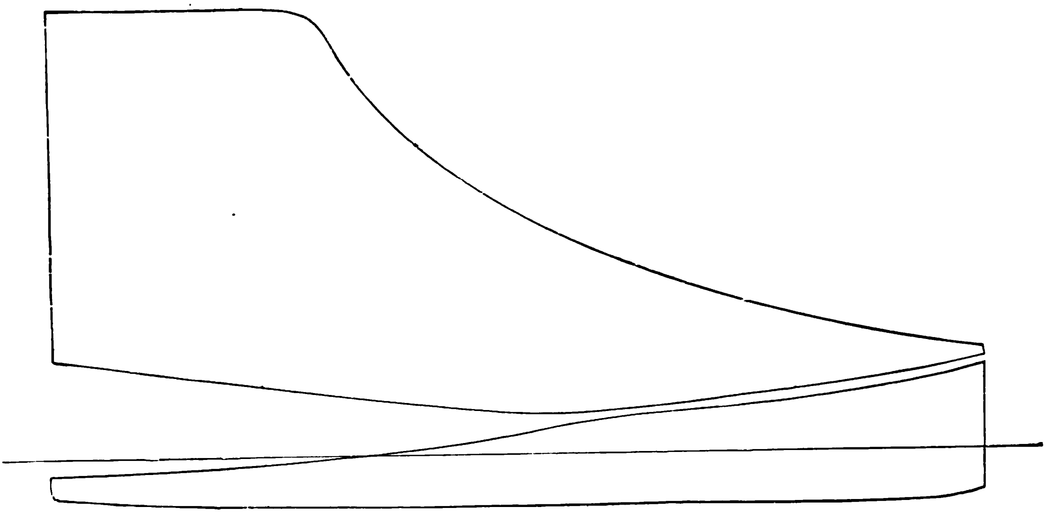
40.	Water evaporated per hour, from temperature of 100° Fahr. into steam of 70 lbs. gauge pressure excluding feed water heater.	per sq. ft. of grate surface, -	63.8 lbs.
41.		per sq. ft. of water heating surface,	3.16 lbs.
41a.		per sq. ft. of superheating surface,	5.38 lbs.
41b.		per sq. ft. of total heating surface,	1.99 lbs.
42.		per sq. ft. of least area for draught,	455.1 lbs

COMMERCIAL HORSE-POWER.

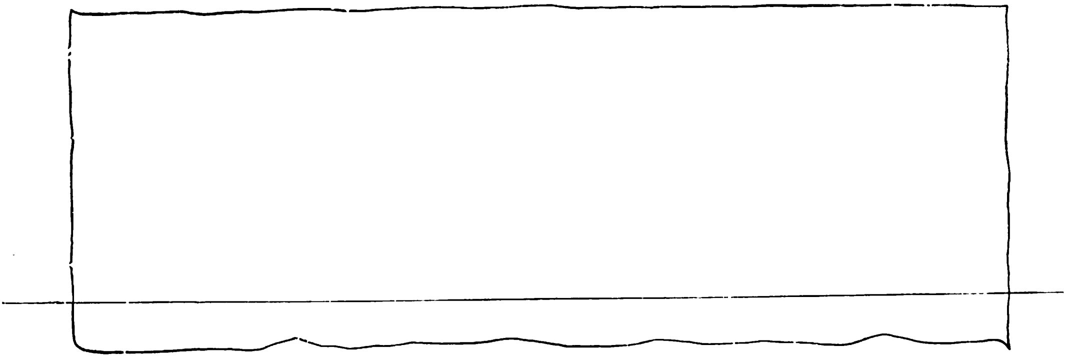
43. On the basis of 30 pounds of water per hour, evaporated from temperature of 100° Fahr. into steam of 70 pounds gauge pressure, (= 34½ pounds from and at 212° Fahr.:)

Including feed water heater, - - 82.1 H. P.
Excluding feed water heater, - - 78.9 "

INDICATOR CARDS TAKEN FROM NEW BEDFORD ENGINE DURING TEST.



STEAM END.



WATER END.

MONTREAL, P. Q.

10,000,000 IMPERIAL GALLONS CAPACITY.

ONE ENGINE.

REPORT OF MESSRS D. KEARNEY AND E. O. CHAMPAGNE

To the Chairman and Members of the Water Committee of the City of Montreal:

GENTLEMEN—We, the undersigned, having been appointed by your honorable Board to test the new Worthington High Duty Pumping Engine, erected in the building known as No. 1 Engine House of the Montreal Water-Works, with the object of ascertaining whether or not the engine comes up to the contract requirements, which are that the engine pump ten (10) millions of imperial gallons of water in twenty-four hours, against a constant reservoir pressure of eighty (80) pounds pressure per square inch, including friction in water mains, with a steam pressure of eighty (80) pounds, and a piston speed of one hundred and ten (110) feet per minute; the city furnishing steam at eighty (80) pounds per square inch, with an evaporative duty of nine (9) pounds of water per pound of coal.

The above engine has two (2) high-pressure cylinders 28¾ inches diameter; two (2) low-pressure cylinders 57½ inches diameter, and two (2) water plungers 31½ inches diameter, and all of 50-inch stroke.

The recent performances of the engine previous to the test, made it clear to your experts that one departure from the letter of the contract would have to be made in order that everything would be definitely settled before the test commenced; your experts and those of the contractors discussed the following points, and settled the same, as follows:

1st. The circular issued calling for tender which are made an essential part of the contract, state: that steam will be furnished by the city at eighty (80) pounds pressure, and that the engine will be required to work at a proper speed against a constant reservoir pressure of eighty (80) pounds per square inch, including friction of mains.

It will readily be seen that the forcing of ten (10) millions additional gallons of water through the same mains, must necessarily raise the pressure which is now found to be, instead of eighty (80) pounds with the new engine working, 92.5 pounds.

In order that this may be fairly met, one of two things had to be done, either to stop some of the hydraulic machinery and lower the water in the reservoir so as to reduce the pressure to the required limits, or allow an increased steam pressure equal to the increased water pressure pound for pound.

The former was difficult and expensive and would be attended with much fluctuation, therefore the steam pressure was allowed.

2d One of the new economical devices in connection with the engine is that the steam used in the cylinder jackets when condensed there is trapped and returned to the boilers, at the temperature of the steam, by a small pump for the purpose. Hitherto this water has been wasted, as is the case with our old Worthington engine.

The quantity of water in pounds returned to the boiler in this manner at the gravity due to its temperature was found to be 21.551 pounds. If this water had not been returned, its equivalent would have to be furnished at the natural temperature of the water.

The contractors claimed that they be allowed the difference in temperature, or, in other words, that they be allowed for the heat they put into the water.

It was allowed.

3d. The contract states that the contractor will be furnished with an evaporation of nine (9) pounds of water per pound of coal, and steam at eighty (80) pounds, they claim that the percentage of water found in the steam be allowed them.

In order that this percentage should be ascertained, calorimeter tests were made every three hours during the trial, the result of which proved that there was 5$\frac{1}{10}$ per cent. of water carried over with the steam.

The claim was allowed.

4th. The temperature of the water in its natural state was found to be 71 degrees, and entering the boiler, 189 degrees, the additional heat being added by means of a special arrangement utilizing the exhaust steam from the auxiliary pumps which form a portion of the engine proper.

The contractor asked to be allowed for the heat put into this water, which was allowed.

5th. The plunger slippage was estimated at 1½ per cent., this being the average slip found by many weir measurements in the United States. Weir measurements in our case would be difficult and expensive.

In submitting our report we do not deem it necessary to enter into a description of the new mechanical arrangements which distinguish this engine from the well-known old Worthington engine, as such has already been fully described to the world in papers, pamphlets and periodicals, which, no doubt, have reached the greater portion of the mechanical engineering profession.

The contract stipulates that the duty is to be ascertained by a special test.

There being no definite length of trial stated in the contract, the Worthington representatives and the undersigned agreed upon a test of twenty-four (24) hours, which was conducted in the following manner:

The fires were lighted at 5 o'clock A. M., steam being raised slowly, the engine was started at 8 o'clock A. M., and got up to full speed at 11 o'clock A. M.

The test commenced at 12 o'clock noon, when a full reading of all gauges was taken, and height of water in the boiler glass gauges taken and noted, the steam, water, and vacuum gauges were read and noted every half hour. Two revolution counters were used, both worked faithfully; the water meter was taken every half hour and the same tested ten times during the trial; complete sets of cards were taken.

The boilers are of the Heine type, two being used to furnish steam to the engine on trial, and the other furnished steam to work the independent boiler feed pump.

The water, steam, and other gauges were tested before and after trial, also the indicators.

The engine worked smoothly and well, a detailed result of the observations will be found in the table annexed to this report.

We have much pleasure in stating that the contractors' representatives acted throughout the trial in a manner that reflects the highest credit upon themselves and the company they represent, manifesting a spirit of fairness that made our duty agreeable.

The whole respectfully submitted,
We have the honor to be, Gentlemen,
Your humble servants,
D. KEARNEY,
ED. OCT. CHAMPAGNE.

MONTREAL, 4th August, 1888.

RESULT OBTAINED ON TEST OF NEW WORTHINGTON ENGINE AT WHEEL HOUSE M. W. W., 30TH AND 31ST OF JULY, 1888.

Test commenced at 12 o'clock noon, 30th July, 1888.
" ended, 31st " "

1.	Duration of test, - - - - - - -	24 hours.
2.	Reading of counter at commencement of test, -	230,158
3.	Reading of counter at end of test, - - -	249,218
4.	Total number of counts made by engine, - -	19,060
5.	Total number of strokes made by engine, - -	76,240
6.	Average length of strokes in feet, 4.165, - -	4.165
7.	Total travel of plunger in feet, (5) × (6), - -	317,539.6
8.	Total travel of plunger in inches, (7) × 12, -	381,047.5
9.	Average area of plunger in square inches, - -	762.21
10.	Total displacement of plunger in cubic inches, (8) × (9), - - - - - - -	2,904,382,127
11.	Total imperial gallons pumped in 24 hours, (10) ÷ 277.274, - - - - - - -	10,471,166
12.	Excess above guarantee, - - - - - -	471,166
13.	Percentage of excess above guarantee, (12) ÷ 10,000,000 × 100 = - - - - -	$4\frac{71}{100}$ per ct.
14.	Deducting percentage of slip = - - -	$1\frac{50}{100}$ "
15.	Net percentage of excess above guarantee, (13) — (14) = - - - - - - - - - -	$3\frac{21}{100}$ "
16.	Average piston speed of engine for 24 hours, = (7) ÷ 2 ÷ 1,440 = - - - - - - -	110 ft.
17.	Average head on plunger, - - - - - -	92.5
18.	Total load on plunger, = (9) × (17) = -	70,504.425 lbs.
19.	Total foot-pounds of work done by engine in 24 hours, = (7) × (18) = - - - -	22,387,946,912.73
20.	Reading of meter counter at commencement of test, - - - - - - - -	13,592
21.	Reading of meter counter at end of test, - -	16,883
22.	Total cubic feet registered by meter, - -	3,291
23.	Weight of one cubic foot of water as determined by testing meter = - - - - - -	65.8
24.	Total weight of water registered by meter in pounds (22) × (23), - - - - - - - - -	216,547.8
25.	Average temperature of water in basin, - -	71 deg.
26.	Average temperature of feed, - - - -	189 deg.

27. British thermal units required to evaporate one pound of water from 71 degrees to steam at 89 pounds = 1,143.7

28. British thermal units required to evaporate one pound of water from 189 degrees to steam at 89 pounds pressure, - - - - - - - - - - 1,025.7

29. Total equivalent pounds of water evaporated at temperature of 71 degrees =(24) ÷ (27) × (28) = 194,205

30. Water wasted in testing meter, - - - - 1,974.5 lbs

31. Water wasted in testing steam = - - - 100 "

32. Total water to be deducted from feed, (30) + (31) = - - - - - - - - 2,074.5 "

33. Jacket water returned without being metered= 14,636.95

34. Net feed water consumption by engine = (29) — (32)+ (33), and deducting for water in steam, - - 196,222.3

35. Coal consumed on basis of 9 pounds of water per pound of coal (34) ÷ 9 = - - - - - 21,802.47

36. Foot-pounds duty of engine per 100 pounds of coal on 9 pounds basis (19) ÷ (35) × 100 = - - 102,685,369

37. Excess of duty above guarantee = - - - 17,685,369

38. Percentage of excess above guarantee (37) ÷ 85,000,000 × 100 = - - - - $20\frac{80}{100}$ per ct.

DAVENPORT, IA.

5,000,000 GALLONS CAPACITY.
ONE ENGINE.

REPORT OF JAMES DONAHUE, VICE-PRES., AND THOMAS HOOPER, SECY.

GUARANTEE.

"The engine will be capable of achieving a duty of 110,000,000 foot-pounds for each 100 pounds of coal consumed, calculated on a basis of 10 pounds of feed water, evaporated from a temperature 70 degrees for every pound of coal burned. The engine to run during this test against a reservoir pressure not exceeding 100 pounds and supplied with steam at a pressure of 100 pounds per square inch."

PRINCIPAL DIMENSIONS.

Type of engine, - - - - - -	Horizontal High Duty.
Diameter of high pressure cylinder, - -	21 in.
Diameter of low pressure cylinder, - -	42 "
Diameter of pump cylinders, - - -	19½ "
Diameter of high pressure rods, - -	3¾ in. and 4¾ "
Diameter of low pressure rods, - -	3¾ "
Diameter of plunger rods, - - -	3½ in. and 4¾ "
Nominal stroke, - - - - -	36 "

RECORD OF TEST.

Date of trial, - - - - - - - -	April 28th, 1888.
Duration of trial, - - - - - -	8 hours
Total counts, - - - - - - - -	9,635
Total strokes, - - - - - - - -	38,540
Average length of strokes, - - - -	37.9 in.
Average strokes per minute, - - - -	80.3

Average displacement of plunger,	46.753 gals.
Average reading of water pressure gauge,	240.26 ft.
Average reading of well gauge,	17.35 ft.
Total head, 1 ft. added for friction,	258.61 ft.
Gallons pumped during trial,	1,801,700
Rate for 24 hours,	5,405,100 gals.
Weight per gallon,	8.34 lbs.
Total feed water,	31,908.27 lbs.
Total coal on 10 to 1 evaporation,	3,190.82 lbs.

$$\text{Duty} = \frac{46.753 \times 8.34 \times 38{,}540 \times 258.61 \times 100}{3{,}190.82} = 121{,}795{,}222 \text{ ft.-lbs.}$$

Average steam pressure,	101.7 lbs.
Average vacuum,	28.2 in.
Average temp. of engine-room,	70 deg.
Average temp. of outside air,	40 “
Average temp. of injection water,	38 “
Average temp. of air pump discharge,	85 “
Average temp. of feed water at entrance to heater,	96.6 “
Average temp. of feed water at exit from heater,	230.7 “
Indicated horse-power,	255.4 “
Efficiency,	96.3 per ct.

HAMPTON, ENGLAND.

20,000,000 IMPERIAL GALLONS CAPACITY.

ONE ENGINE.

REPORT BY PROFESSOR UNWIN, F. R. S.

First Report on Trials of a Worthington High-Duty Pumping Engine at the West Middlesex Water Works at Hampton.*

Acting on instructions from Messrs. Simpson & Co., I carried out complete trials of the new Worthington engines at Hampton. In making the observations, I was assisted by Mr. Taylor, by some of my students, and by some of Messrs. Simpson's pupils. Nearly all observations were made in duplicate by two independent observers.

In consultation with Mr. Mair, I had previously arranged as to the preparations to be made for the trials, and I have to acknowledge the care with which every provision required was made. All the information needed in carrying out the trial and discussing the results has been unreservedly placed at my disposal. Mr. M. W. Hervey, the engineer of the West Middlesex Water Works, and his assistant were good enough to facilitate in every way the trial. It was arranged that there should be an eight hours' trial of the engines only, and a twenty-four hours' trial of engines and boilers conjointly. Messrs. Simpson sent down a supply of Welsh coal for the purposes of the trial. They also provided feed-measuring tanks and a tank for gauging the air-pump discharge. The jacket drains were re-arranged so that the jacket condensation could be measured. In every respect my wishes were attended to, and Messrs. James Simpson & Co. were unremitting in care to make the trial as complete as possible.

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* This engine was illustrated in *Engineering*, January 4, 1889.

The Engines.—The engines are compound high-duty Worthington engines pumping a very large volume of water on a comparatively low lift. The high-pressure pistons are 27 inches in diameter, and the low-pressure pistons 54 inches. The stroke is variable, the maximum from cylinder head to cylinder head being 44 inches. During the trials the stroke remained very constant and about 43 inches. The valves of one engine are worked by the other engine, but arrangements are made insuring fixed points of cut-off in each cylinder, and independent control of the compression in the high-pressure cylinder. The engines work directly double-acting ram pumps, with rams 40 inches in diameter, and of course having the same stroke as the steam pistons. The pump valves are of India-rubber, spring loaded, and in all probability the slip is extremely small. The peculiarity of the engine is that there are compensating cylinders which absorb work during the first half of the stroke, and give it back during the second half. There are two of these compensators to each engine, 11 inches in diameter, loaded by air-pressure to about 120 pounds per square inch. The pumps lift water from a well communicating with the river, and deliver it through two 3-foot mains to the reservoirs at Barnes, about 9 miles distant. The head during the trials, measured by the difference of pressure in the suction and discharge pipes, was from 50 feet to 65 feet, a head almost entirely expended in overcoming the friction of the main. The head was measured by mercury columns fixed in the engine house, communicating with the suction and delivery mains in accordance with the provisions of the contract.

The suction gauge communicated with the suction pipe just below the floor, and the pressure gauge was set to give pressures reckoned from the floor level. The sum of the mercury gauge readings is taken as the effective lift. It should be noted, however, that the pressure gauge communicates with the delivery main at a point beyond the stop-back valve. Consequently the resistance of that valve is reckoned as part of the engine friction, and is not credited to the useful work done by the pumps.

The engine cylinders are completely jacketed, and the steam is also taken through a jacketed reservoir between the cylinders. The jacket water was discharged through a pipe regulated by a stop valve and weighed. The condensers are injection condensers with horizontal air pumps.

Before the trials, I attended at Hampton and tried some gauges, which had been prepared, of the diameters of cylinders, pumps and rods. These were then very carefully measured, with the following results:

DIAMETER AND AREAS OF CYLINDERS AND PUMPS.

		Diameter at 60 deg. F.	Diameter at 316 deg. F.	Area of Piston.	Area of Rod.	Effective Area.	Means.
		in.	in.	sq. in.	sq. in.	sq. in.	
H. P. cylinder A.	Back	26.98	27.02	573.4	17.7	555.7	553.5
	Front	26.98	27.02	573.4	23.8	549.6	
H. P. cylinder B.	Back	27.02	27.06	575.1	17.7	557.4	
	Front	27.02	27.06	575.1	23.8	551.3	
L. P. cylinder A.	Back	53.99	54.07	2296.2	7.0	2289.2	2285.1
	Front	53.99	54.07	2296.2	17.7	2278.5	
L. P. cylinder B.	Back	54.02	54.10	2298.7	7.0	2291.7	
	Front	54.02	54.10	2298.7	17.7	2281.0	
Pump plungers	Back	39.90	....	1250.0	16.8	1233.2	1241.6
	Front	39.90	....	1250.0	0	1250.0	

The Boilers.—The boilers are single flued Cornish boilers. Three were used in the trial on October 29th, and four in the trial on November 5th and 6th. The boilers are 28 feet in length and 6 feet in diameter, with a single flue 3 feet 6 inches in diameter for the greater part of the length.

During the trials of November 5th and 6th the length of the grate was 4 feet 6 inches. Hence the grate area of the four boilers was 63 square feet. During the trials there was no leak from the blow-offs of the boilers in use, or from the boilers in use into those standing idle. The feed pipe was disconnected, and the safety valves open on the idle boilers.

The coal was weighed under supervision on platform scales which had been tested, and the weights of coal brought into the house were from time to time again tested on a Denison balance, which I had taken down for that purpose, and which was in perfect adjustment.

Measurement of the Feed.—The feed was supplied from the delivery main of the pumps at a nearly constant temperature, the ordinary feed arrangements which supply the boilers with hot water from the jackets and hot-well being disconnected. The feed was delivered into a small open top gauge tank with overflow pipe provided with a float and counter.

The inlet cock, outlet cock and overflow pipe of the feed gauging tank all discharged in the air in sight, so that it was practically impossible to make a mistake in filling and emptying the gauge tank.

The capacity of this gauge tank was determined three times by weighing the water; and the closely accordant measurements gave a mean value of 394 pounds for the capacity. As these measurements were made with the actual feed water and with the tank in place and undisturbed, no corrections are necessary for temperature and no error is introduced by any possible difference of level or condition.

The feed gauging tank delivered by a stop valve into another tank from which a small Worthington feed pump delivered the water into the boilers.

The Worthington pump took its steam from the boilers in use and exhausted into a coil in the feed tank, from which it pumped. The whole of the steam used by the Worthington feed pump was therefore recondensed and returned to the boilers.

Of the heat supplied by the boilers to work the feed pump nearly all was returned to the boilers. A small portion—viz., that due to the useful work of pumping and that lost by radiation from the tank, was no doubt lost. So far a small error telling against the main engines is introduced.

The water level at the commencement of each trial in the boiler gauge glasses was carefully observed and the water level was brought to exactly the same marks at the end of the trials. Hence no correction has to be made for difference of level in the boilers. The time at which each tankful was supplied to the boilers was noted, and also the feed water temperature. Pyrometer observations were made in the flues with two Murries pyrometers every quarter of an hour. During part of the trial one of these had to be used at temperatures near the bottom of its scale, where the indications are least trustworthy. But the mean of the pyrometer readings is probably not very incorrect. Anemometer observations of the air supplied to each boiler were taken every half-hour during the twenty-four hours' trial, the anemometer having been previously tested.

Measurement of the Air Pump Discharge.—The air pump discharge was led into a wooden tank with stilling screens. From this it was discharged through one of Mr. Mair's sharp-edged circular orifices freely into the air. The diameter of the orifice was carefully tested after the trials, and the co-efficient of discharge from similar orifices is known to be 0.599. The temperature and head over the orifice were noted every five minutes in the first trial and every seven and a-half minutes in the second. The temperatures relied on in this report were taken by a fixed zero thermometer, with open scale, recently verified at Kew. Observations were also taken by a thermometer of Mr. Mair's, and the means of the two sets of observations differ only by a small amount.

Measurement of Length of Stroke.—As the stroke is variable an arrangement of indicating fingers was attached to each engine, and the length of stroke of each engine was noted every quarter of an hour.

Indicated Power.—The indicated power was taken by four Richard's indicators belonging to Mr. Mair, chosen because they give fairly large diagrams. These indicators were sent to Kensington after the trials and tested under steam, against a steel tube pressure gauge recently made for me and specially tested by Messrs. Schäffer and Budenberg. No important error was found at any part of the scale with any of the springs. But with the light springs of the low-pressure cylinder indicators there was a little frictional sticking or else a little slackness of the parallel motion joints, which under a steady pressure introduced a small uncertainty of indication at one or two points in the range. Probably this would be less still when the indicator piston was in motion as when drawing a diagram. The indicator pipes were large and were clothed. Diagrams were taken every half-hour from all the cylinders, so that there were 128 single diagrams in the eight hours' trial and 384 in the 24 hours' trial. All the eight hours' trial diagrams were reduced by planimeter. Also all the diagrams taken in the first eight hours of the 24 hours' trial, and half those taken subsequently. The conditions were so constant throughout the trial and the diagrams, Figs. 1 and 2, so similar that this was thought sufficient.

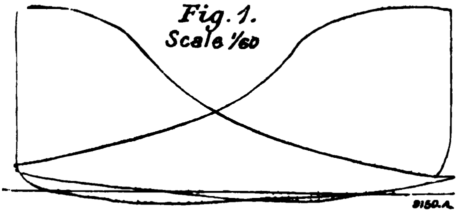
Fig. 1. Scale 1/60

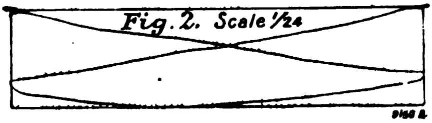
Fig. 2. Scale 1/24

EIGHT HOURS' TRIAL OF ENGINES ON OCTOBER 29.

This was an eight hours' trial, during which the coal used was not noted. The engine was started early in the morning, and the trial began at 11.43 A. M. The mean barometer during the day was 30.166 inches (corrected), corresponding to 14.82 pounds per square inch. The temperature of the injection water taken in the well outside the engine-house was almost constant at 54° Fahr. The mean boiler pressure during the trial was 75.2 pounds per square inch (90.02 pounds per square inch absolute), and was kept very nearly constant. The mercurial vacuum gauge on the engine gave a mean vacuum of 28.04 inches or 13.77 pounds per square inch.

The head of water on the pumps was 63.75 feet at starting, and diminished to 59.9 feet at the end of the trial, the diminution being tolerably regular and the variations of head small. No doubt the variation of head depends chiefly on the engine speed, but it also depends on the levels at the discharging reservoir. The mean head during the trial was 60.63 feet. The air pressure in the compensating air vessel varied very little, only a pound or two during the whole trial; the mean pressure was 120.6 pounds per square inch (above atmosphere).

Speed and Length of Stroke.—The speed was remarkably constant throughout the trial. The greatest variation from the mean speed, from observations of the counter every quarter of an hour, did not exceed half of 1 per cent. of the mean speed. The mean speed for the whole trial was 17.67 double strokes per minute, the engines making altogether 8,480 double strokes.

The length of stroke was also extremely constant, and this is remarkable, since the length of stroke depends directly on the adjustment of the expansion and compensating air pressure.

The minimum stroke observed was 41.96 inches, and the maximum 43.56 inches.

The mean stroke during the trial was 42.83 inches for engine A, and 43.00 inches for engine B. Mean for both engines, 3,576 feet.

Indicated Horse-Power.—The reduction of the half-hourly sets of diagrams gives the following results:

	Indicated Horse-Power.		
Engine A.—H. P. back, - - - -	37.78		
" L. P. " - - - -	34.19	71.97	143.34
" H. P. front, - - - -	38.98		
" L. P. " - - - -	32.39	71.37	

	Indicated Horse-Power.		
Engine B.—H. P. back,	45.68		
" L. P. "	32.13	77.81	152.91
" H. P. front,	43.93		
" L. P. "	31.17	75.10	
Total indicated horse-power, both engines,			296.25

The Pumps.—The mean lift was 60.63 feet, and the number of strokes, 3.576 feet in length, was 17.67 per minute. Hence the pumps lifted 13,598 gallons per minute, or 815,862 gallons per hour, or at the rate of more than 19,580,000 gallons per day, no slip being allowed for. The pump horse-power is 249.84, and consequently the mechanical efficiency of the engine and pumps is 0.8434, a very high efficiency when the low lift is considered.

The Feed Water and Jacket Water.—The feed water, measured as described, had a mean temperature of 55° Fahr. The total quantity used in the eight hours was 41,277.8 pounds, or 5159.7 pounds per hour. The total quantity of jacket water was 5357 pounds, or 669.6 pounds per hour. Consequently, reckoned per indicated horse-power per hour, the quantities used were:

	Pounds per I. H. P. per hour.
Total feed (at 55 degrees) per indicated horse-power per hour,	17.41
Jacket water ditto,	2.26
Used in cylinders,	15.15

Air Pump Discharge.—The mean head over the orifice was 1.9662 feet, and the mean temperature was 81.18° Fahr.[*] Taking one cubic foot of water at 81 degrees to weigh 62.2 pounds, the total air pump discharge was 2777.7 pounds per minute, or 2702.9 pounds of injection water and 74.8 pounds of condensed steam.

Heat Rejected by the Engine per Indicated Horse-Power per Minute.—If we take the heat rejected as being approximately that required to raise the total air-pump discharge from 54 degrees to 81.18 degrees, we get for the heat rejected 254.8 thermal units per indicated horse-power per minute This is the quantity ordinarily known as Donkin's co-efficient. As, however, here the indicated horse-power and the feed and jacket discharge were measured, a nearer approximation can be obtained.

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[*] Observations by another observer with another thermometer made this temperature 81.52 degrees. If this result is adopted the heat unaccounted for is less by about 3.2 thermal units per indicated horse-power per minute.

	Thermal Units per minute.
Heat due to 2702.9 pounds of injection per minute raised 27.18° Fahr.,	73,460
Heat due to 74.8 pounds of feed per minute raised 26.18 degrees,	1,958
Heat due to 11.16 pounds of jacket water per minute raised 265 degrees,	2,958
Total,	78,376
Heat rejected per indicated horse-power per minute,	264.6
Add converted into work,	42.7
Total,	307.3

This is the total heat used by the engine, neglecting radiation and similar losses.

Heat Used Reckoned from the Boiler Steam.—The total heat of steam reckoned from the feed temperature 55 degrees at the mean boiler pressure is 1156.4 thermal units per pound. Consequently the heat delivered from the boiler to the engine, if we assume the steam dry and free from priming water, was 335.5 thermal units per indicated horse-power per minute. The difference between this and the previous estimate of 307.3 thermal units represents loss by radiation, error due to the presence of priming water, and errors of observation.

If we suppose the jacket water pumped back into the boiler at the temperature of the steam (as it returns in a closed circuit), and the rest of the feed taken from the hot-well at 81 degrees, conditions which occur in the ordinary working of the engines, then 15.5 thermal units, or 4.62 per cent. of the heat used, would be saved, which in the abnormal conditions required for the purposes of the trial were wasted. Consequently the total heat used per indicated horse-power per minute would be 320 thermal units.

The following table gives the results tabulated in the same way as in Mr. Mair's paper on a "Direct-Acting Steam Pump" (Minutes of Proceedings Inst. C. E., Vol. LXXXVI.):

Double stroke per minute,	17.67
Boiler pressure,	75.2 lb. per sq. in.
Feed water per minute,	85.99 lb.

Jacket drains per minute,	11.16 lb.
Temperature of steam,	320.06 deg.
Pressure on pump,	26.27 lb. per sq. in.
“ in compensators,	120.6 “ “
Mean pressure in H. P. cylinders,	39.23 “ “
“ L. P. “	7.42 “ “
Temperature of injection,	54° Fahr.
“ air-pump discharge,	81.18° “
Head over orifice,	1.9662 ft.
Air-pump discharge per minute,	2777.7 lb.
Injection water,	2702.9 “

Heat Passing through Engine per Minute per Indicated Horse-Power.

Thermal units from boiler in saturated steam through cylinders from feed temperature,	292.0
Latent heat of jacket steam,	33.6
	325.6
Heat rejected in air-pump discharge,	254.6
Converted into work,	42.7
Radiation and error,	28.3
	325.6
Indicated horse-power,	296.25
Pump horse-power,	249.84
Mechanical efficiency,	.8434
Feed per indicated horse-power per hour through cylinders,	15.15 lb.
Feed per indicated horse-power per hour through jackets,	2.26 “
Piston speed per minute,	126.4 ft.

TWENTY-FOUR HOURS' TRIAL OF ENGINES ON NOVEMBER 5 AND 6.

This was a twenty-four hours' trial, the coal consumption being measured as well as the efficiency of the engines. The engines, as before, had been started in the morning, but before beginning the fires were cleaned and all ashes removed; also all coal was swept from the boiler-house floor. Four boilers were used, and the fires were not drawn; but the condition of the fires was nearly identical at the beginning and end of the experiment. The fires were cleaned again about eighteen hours after starting, all the clinker and ash removed being placed in the ash-pits. At the end of the trial I judged the fires to be on the average slightly thicker than at the beginning of the trial. The trial commenced at 10.22 A. M. on the 5th, and ended exactly at 10.22 A. M. on the 6th.

The barometer varied a little during the twenty-four hours, the mean being 29.78 inches (corrected), corresponding to 14.627 pounds per square inch. The temperature of the injection varied from 48.6° Fahr. to 49.5° Fahr., the mean being 49.2° Fahr. The mean boiler pressure was 60.29 pounds per square inch (74.92 pounds per square inch absolute). The mean vacuum shown by the mercury gauge on the engine was 27.76 inches, or 13.63 pounds per square inch. The total head of water on the pumps was about 55 feet at starting and 53.5 feet at the end of the trial. It varied little during the trial, and the mean head was 53,68 feet. The air pressure in the compensating air vessel varied from 118 pounds to 122 pounds per square inch (above atmosphere).

Speed and Length of Stroke.—As in the previous trial the speed was remarkably constant, the mean speed being 17.282 double strokes per minute. The engines made 24,886 double strokes in the twenty-four hours. The length of stroke was even more constant than in the previous trial. The extremes were 42.32 inches and 43.56 inches. The mean length of stroke was 43.06 inches for engine A, and 43.05 inches for engine B. Mean for both engines, 3.5879 feet.

Indicated Horse-Power.—The reduction of diagrams taken every half hour during the first eight hours, and every hour afterwards, gave the following results. The variation of the diagrams was very small.

	Indicated Horse-Power.		
Engine A.—H. P. back,	31.662		128.668
" L. P. "	31.145	62.807	
" H. P. front,	34.176		
" L. P. "	31.685	65.861	
Engine B.—H. P. back,	35.856		126.849
" L. P. "	28.073	63.929	
" H. P front,	35,236		
" L. P. "	27.684	62.920	
Total indicated horse-power of both engines,			255.517

The Pumps.—The mean lift was 53.68 feet, mean length of stroke, 3.5879 feet. Number of strokes per minute, 17.282. Hence the pumps lifted 13,407 gallons per minute, or 804,396 gallons per hour, or 19,305,504 gallons in the twenty-four hours. The pump horse-power is 217.06. Consequently the mechanical efficiency of the engines and pumps is 0.8495, slightly greater than in the previous trial. This may be due to a slight readjustment of the valves between the trials which made the work of the two engines more nearly equal.

The Feed and Jacket Water.—The feed water had a mean temperature of 51.07 degrees. The total feed water used was 108,537.4 pounds, or 4522.39 pounds per hour. The jacket water was measured for six hours on the 5th, and for one hour on the morning of the 6th. The rate of discharge appeared to be the same. The amount of drainage from the jackets was 706 pounds per hour. Consequently reckoned per indicated horse-power per hour the quantities were:

	Pounds per I. H. P. per hour.
Total feed (at 51.07 deg.) per indicated horse-power per hour, - - - - - - - - -	17.700
Jacket condensation, - - - - - - -	2.763
Used in cylinders, - - - - - -	14.937

Air-Pump Discharge.—The mean head over the orifice was 1.7033 feet, and the mean temperature 74.965 degrees.* The total air-pump discharge was 2,586 pounds per minute, or 2522.4 pounds of injection water, and 63.6 pounds of condensed steam.

Heat Rejected by the Engine per Indicated Horse-Power per Minute.—Taking the heat required to raise the whole air-pump discharge from 49.2 degrees to 74.965 degrees, we get for the heat rejected 260.7 thermal units per indicated horse-power per minute. This is Donkin's co-efficient. The more accurate estimate of the heat rejected, as in the previous trial, is as follows:

	Thermal Units.
Heat due to 2522.4 pounds of injection water per minute raised from 49.2° Fahr. to 74.965° Fahr.,	64,990
Heat due to 63.6 pounds of feed water raised from 51.07° Fahr. to 74.965° Fahr., - - - -	1,519
Heat due to 11.78 pounds of jacket water raised 256.3° Fahr., - - - - - - - -	3,020
	69,529

---

* Another set of readings with another thermometer gave this temperature 75.13° Fahr.

Heat rejected per indicated horse-power per minute,	272.1
Add, converted into work, - - - - - -	42.7
Total, - - - - - - - -	314.8

This is the total heat used, neglecting the loss by radiation.

Heat Used Reckoned from the Boiler Pressure.—The total heat of the steam, considered dry, reckoned from the feed temperature at the mean boiler pressure is 1156.5 thermal units per pound. Consequently the heat delivered from the boiler to the engine was 341.1 thermal units per indicated horse-power per minute. The difference between this and the previous estimate of 314.8 represents loss from radiation, error due to the presence of priming water in the steam, and errors of observation.

If, as before, we suppose the jacket water pumped into the boiler at the temperature of the steam (as it returns to boilers in a closed circuit), and the rest of the feed taken from the hot-well, thus removing the abnormal conditions which were present in the trial, 17.8 thermal units, or 5.2 per cent. of the heat used per indicated horse-power per minute would be saved. Then the heat required by the engine would be in normal conditions of working 323.3 thermal units per indicated horse-power per minute. This is slightly greater than in the previous trial, as would be expected from the lower boiler pressure and slower speed.

The following table gives the results tabulated in the same way as before:

Double strokes per minute, - - - -	17.282
Boiler pressure, - - - - - - -	60.29 lb. per sq. in.
Feed water per minute, - - - - - -	75.37 lb.
Jacket drains per minute, - - - -	11.77 "
Temperature of steam, - - - - - -	307.36° Fahr.
Pressure on pump, - - - - - - -	23.26 lb. per sq. in.
" in compensators, - - - -	120 " "
Mean pressure in high-pressure cylinders, -	32.92 " "
" low-pressure " - -	6.905 " "
Temperature of injection, - - - -	49.2° Fahr.
" air-pump discharge, - -	74.965° "
Head over orifice, - - - - - - -	1.7033 ft.
Air-pump discharge per minute, - - -	2,586 lb.
Injection water per minute, - - - -	2522.4 "

Heat passing through engine per indicated horse-power per minute:

Thermal units from boiler in saturated steam through cylinders from feed temperature,	287.8
Latent heat of jacket steam,	41.45
	329.25
Heat rejected in air-pump discharge,	260.24
Converted into work,	42.75
Radiation and error,	26.26
	329.25
Indicated horse-power,	255.517
Pump “	217.06
Mechanical efficiency,	.8495
Feed per indicated horse-power per hour through cylinders,	14.937 lb.
Feed per indicated horse-power through jackets,	2.763
Piston speed per minute,	124 ft.

It should be noted here that the engines worked for twenty-four hours with the greatest regularity of speed and stroke, and this although the steam and expansion valves remained untouched throughout the trial after their first adjustment at starting.

THE BOILERS.

Measurement of Coal Used.—The stoke-hole floor having been swept clean at the beginning of the trial, the coal was brought in in quantities of about 8 cwt., and the time of finishing each lot was noted. The ash-pits were cleaned before the trial, and afterwards nothing was removed till the end of the trial. The fires were cleaned before the trial began, and again at 4 A. M. on Tuesday morning. The fires were not touched at the end of the trial, but the ash-pits were immediately cleaned, and the whole of the ashes were treated thus:

First the clinkers, including those removed from the fires at 4 A. M. (six hours before the end of the trial), were separated and weighed. The rest of the ashes were sifted through a sieve with ½-inch mesh. All that passed through the sieve is treated as incombustible ash, although probably one-third of it is unburned carbon. What did not pass through the sieve is treated as unburned fuel. Analysis in similar cases has shown that the cinders retained by the sieve are almost entirely carbon.

The coal account then stands thus:

	lb.	lb.
Gross weight of coal brought into boiler house, -		11,180
Left on floor at end of trial, - - - - - -	99	
Cinders sifted out of ashes, - - - - - -	132	231
Total coal used - - - - - - -		10,949
		= 456.2 lb per hour.

	lb.
The residue consisted of clinkers, - - - -	66
Incombustible ashes, - - - - - - -	366
	432

The clinkers and ashes which are reckoned as incombustible amount to 3.9 per cent. of the coal used.

The rate of combustion was 7.24 pounds of coal per square foot of grate, or 0.19 pound per square foot of heating surface per hour. The coal used per indicated horse-power per hour was 1.785 pounds, a very good result, as the feed was supplied at 51° Fahr. and the rejected heat from the jacket drains wasted. The evaporation was 9.914 pounds of water from 51.07 degrees at 307.36 degrees per pound of coal, including clinkers and ashes. This corresponds to an evaporation of 11.867 pounds per pound of coal from and at 212 degrees, no deduction being made for clinkers and ashes.

Calorimetric Value of the Coal.—The heating power of the coal has not been directly determined, but good Welsh coal is known to contain about 89 per cent. of carbon and 4 per cent. of hydrogen, the rest being oxygen, nitrogen and ash. The colorimetric value of such a fuel is

$$14{,}500 \left\{ 0.89 + 4.28 \times .04 \right\}$$
$$= 15{,}387 \text{ thermal units per pound.}$$

But this is reckoned for a dried sample of coal and makes no allowance for the latent heat of the steam produced in combustion. There would be produced by combustion 0.36 pounds of water per pound of coal, and the latent heat of this would be 348 thermal units, so that the available heat of a pound of dry coal would be 15,039 thermal units. The coal as taken from the yard would contain at least 1 per cent. of moisture, so that the available heat of 1 pound of the coal as weighed and used would be:

	Thermal Units.
Heat due to 0.99 pounds of coal, - - - -	14,888
Less latent heat of 0.01 pounds of water, - - -	10
	14,878

Available heat 14,878 thermal units per pound of coal as weighed and used. Taking this value, the total heat due to the combustion of the coal is 26,557 thermal units per indicated horse-power per hour, or 442.6 thermal units per minute per indicated horse-power. Of this 341.1 has been shown to be delivered to the steam. There remains 101.5 thermal units per indicated horse-power per minute to account for as losses in the boilers. The efficiency of the boilers is 0.77. The coal gave to the steam 11,466 thermal units per pound of coal used.

Anemometer Observations.—Observations at each boiler every half-hour gave the following volumes of air entering per minute in cubic feet at the temperature 79.5 degrees of the boiler-house.

Boiler,	J	K	L	M
Quantity of air in cubic feet per minute,	420	438	486	360

Hence, the total quantity of air used was 1,704 cubic feet per minute, or 225 cubic feet per pound of coal. The weight of the air used was 7,489 pounds per hour, or 16.42 pounds per pound of coal. As the coal requires nearly 12 pounds per pound for perfect combustion, the quantity of air used was moderate.

The mean temperature of the flue from the pyrometer observations was 422° Fahr.

Tabulating the results stated, we get:

	per Hour. lb.	Per I. H. P. per hour. lb.
Coal used,	456.2	1.785
Air used,	7489.0	29.310
	7945.2	
Less ashes and clinkers,	18.0	
Total weight of furnace gases,	7927.2	31.03

Heat Used and Lost in Boilers.—The thermal units of heat developed in the furnaces were applied thus:

	Thermal Units per I. H. P. per Hour.	Per Cent.
Total heat due to coal used,	26,557	100
Given to steam,	20,466	77.1
Carried off in furnace gases,	2,657	10.0
Probable loss due to opening fire-doors to stoke,	265	1.0
Due to carbon in ashes,	284	1.1
Radiation and unaccounted for,	2,885	10.8

This calculation depends on an assumption of the calorific value of the coal, but this cannot be far wrong. It assumes that the steam supplied to the engines was dry. If there was any priming water, the heat given to steam would be less. On the other hand, probably the losses due to moisture in the coal, to unburned carbon in the ashes, and to air entering the furnaces during stoking, are underestimated.

Duty of the Engines.—The work done by the engines during the twenty-four hours' trial was 106,010,000 foot-pounds per 112 pounds of coal. Supposing that in the eight hours' trial the heat received by the steam was 11,466 thermal units per pound of coal, as in the twenty-four hours' trial, a supposition which is probably quite exact, then the duty of the engines in the first day's trial must have been 106,513,000 per 112 pounds of coal.

Now it has already been stated that for the purposes of the trial the ordinary conditions of the engines were altered and heat rejected which is ordinarily used. Correcting for this, the duty of the engines in normal conditions of work must be 111.5 millions according to the results of the twenty-four hours' trial, and 111.5 millions according to the results of the eight hours' trial. Figures 1 and 2 are reduced copies of the pairs of high-pressure and low-pressure diagrams.

To accompany this report drawings are sent as follows:

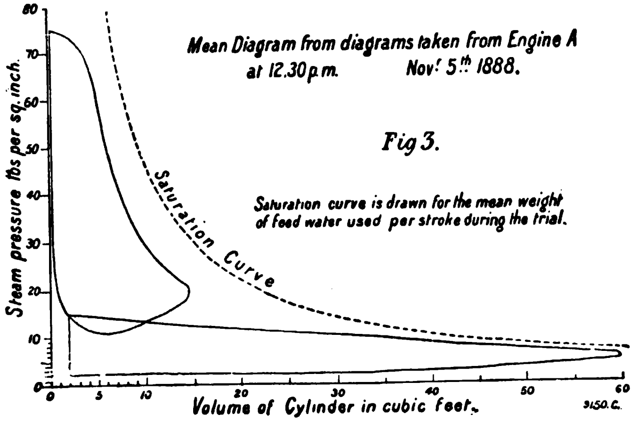

Mean Diagram from diagrams taken from Engine A
at 12.30 p.m. Novr 5th 1888.

Fig 3.

Saturation curve is drawn for the mean weight of feed water used per stroke during the trial.

Drawing 1.—A mean diagram, Fig. 3, drawn from the diagrams taken on engine A at 12.30 P. M. On this has been plotted a saturation curve for the mean speed per stroke during the trial.

Since the indicated power varied so little, this saturation curve must be very approximately the true curve for the actual diagrams. The re-evaporation during the stroke is very marked, as was to be expected from the large jacket condensation.

Drawing 2.—Mean diagrams, Fig. 4, from all the diagrams of both engines taken at 12.30 P. M. are plotted so as to show the effective thrust of the engines at each point of the stroke. The high-pressure cylinder pressures are set up from the base line, and the corresponding low-pressure pressures above these. The pressures were, of course, measured between the forward stroke line of one end and the backward line of the other end of the cylinders to get the true pressures. A curve of cosines is drawn giving the ± thrust of the compensators. Combining this with the engine diagram the resultant thrust is obtained. This is the vertical distance between the low-pressure line and the compensating cylinder line or curve of cosines. These distances set up from the base line give the curve of resultant thrust of engines. The effect of the inertia, which is not large at the low speed of these engines, and the friction of the compensator rams are neglected. It will be seen that the resultant thrust is remarkably uniform, and probably the effect of the inertia of the moving pistons and the friction of the plungers is to increase the uniformity of this thrust.

Drawing 3.—The principal observations taken during the trial have been plotted in this diagram, Fig. 5, with time abscissæ. The diagram shows the general regularity of the working of the engines during the trial.

W. C. UNWIN.

CENTRAL INSTITUTION, SOUTH KENSINGTON,
November 27th, 1888.

DRAWINGS 2 AND 3.—REPORT OF W. C. UNWIN ON TEST OF HAMPTON ENGINE.

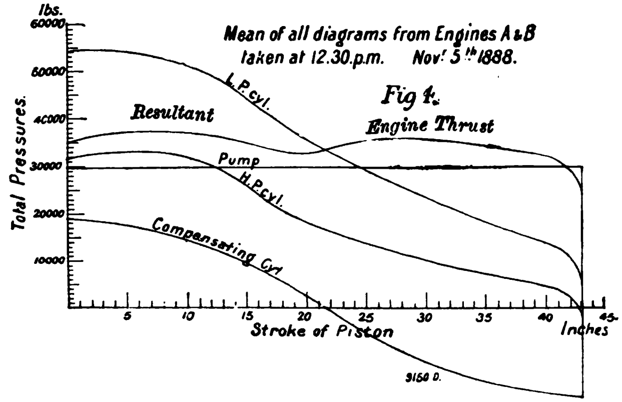


Fig 4. Engine Thrust

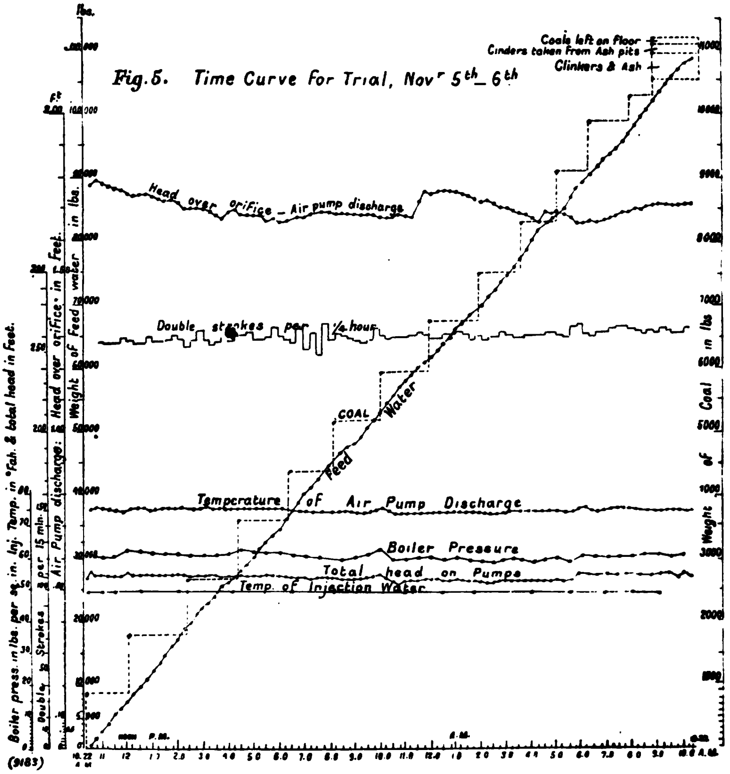
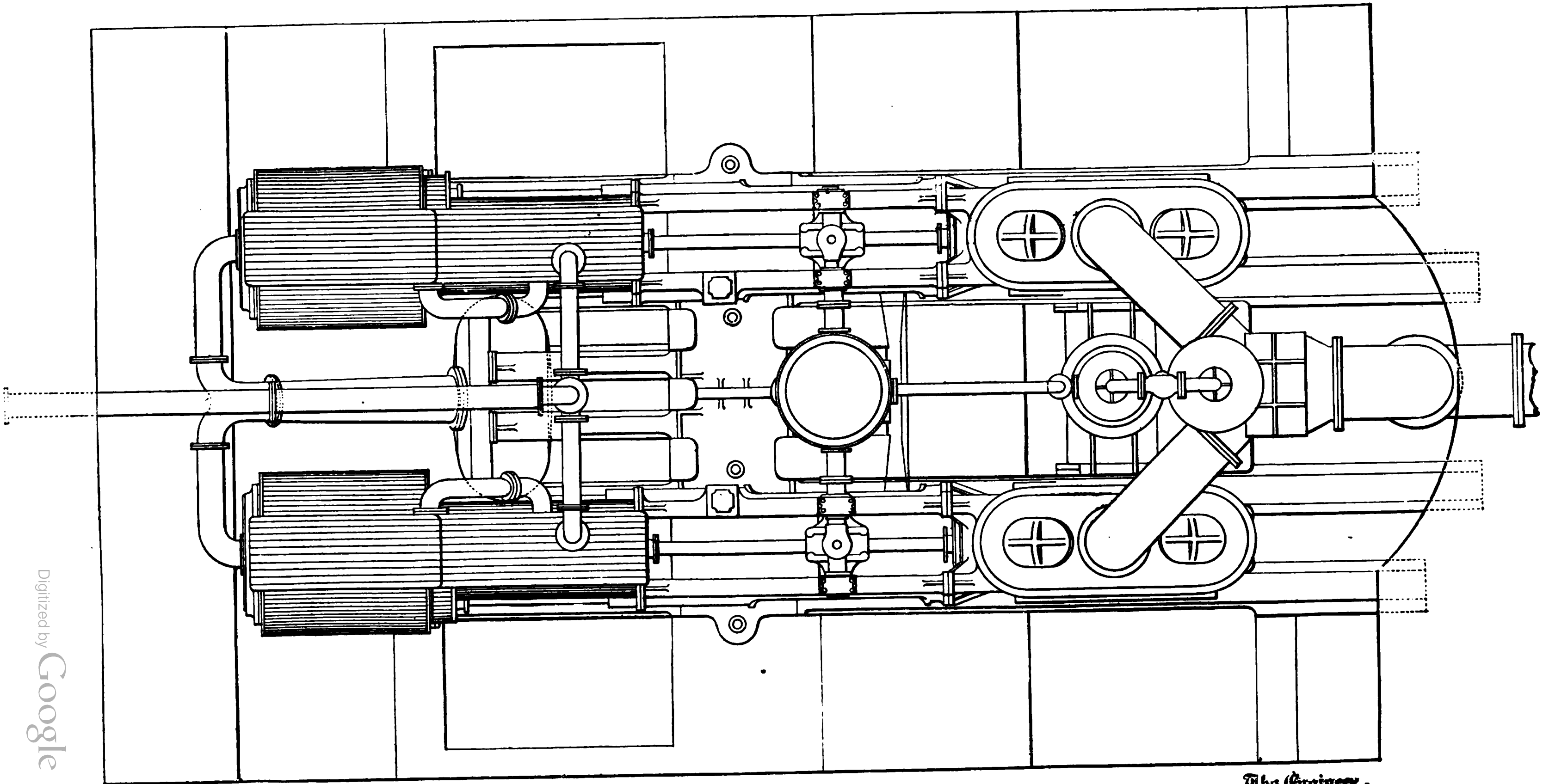


Fig. 5. Time Curve For Trial, Novr 5th_6th



John Swain. Eng.

The Engineer.

LONDON, ENGLAND.

9,000,000 IMPERIAL GALLONS CAPACITY.

ONE ENGINE.

REPORT OF E. A. COWPER, M. INST. C. E.

6 Great George Street,
Westminster, S. W., December 7, 1888.

Messrs. J. Simpson & Co., Limited:

Dear Sirs—I beg to report that, in accordance with your request, I have made a careful trial of the Worthington pumping engine erected at the works of the New River Company, Green Lanes, Stoke Newington, and I have to thank Mr. E. L. Morris, A. M. I. C. E., engineer to the company, for the care he took to insure the trial being as complete as possible.

I arranged for the trial to last 24 hours—viz., from the morning of November 23d till the morning of November 24th. I purposely had the engine and boilers at work for some time, in an entirely normal condition, before commencing the trial; the state of the fires being particularly noticed both before starting and at the end of the trial, and measured as closely as possible by inserting a brick edgeways on the bars.

A tank holding an exact weight of water was provided, in which the weight of water delivered to the boilers was exactly taken, its temperature being also noted.

The coal, which was "Nixon's navigation," was carefully weighed in scales in the stoke-hole, as required by the boilers.

Indicator figures were taken from each end of each of the four cylinders of the engine throughout the trial. The temperature of the hot well and injection water was taken frequently, as well as the steam pressure, vacuum, and air pressure in the compensating cylinders. The strokes of the engine were registered by a counter, and the head of water in the main and suction was continually noted, and generally every precaution was taken to insure an accurate result as to the working, economy and efficiency of the engine and pumps.

In my opinion the trial was under one disadvantage—viz., that three boilers only were used in place of four, resulting in the heating surface being smaller than one would have wished, in order to obtain the greatest economy. The boilers were Lancashire, or two-flued, and had no Galloway tubes in them.

The engine is of the Worthington horizontal type, there being a steam jacketed high and low pressure cylinder with one piston-rod on each side of the engine, the piston-rod on one side working the steam valves of the other side, so that the motions of the piston-rods follow after one another, and are not simultaneous, except just as one finishes its stroke and the other begins, hence the flow of water in the main is almost exactly uniform and continuous.

The pressure of steam being 80 pounds and the expansion being about twelve times, it was highly satisfactory to observe how completely the Worthington "compensating cylinders" took up the excess of power during the first half of the stroke, and gave it out again during the second half, thus completely avoiding the necessity for a crank and fly-wheel, the velocity of the piston being very uniform throughout the stroke, and starting and stopping so quietly and easily that there was no blow of any kind; in fact, it was almost impossible to tell from the pumps whether they were at work or not. The pump-plungers of the pumps, which are double-acting, are attached to the front ends of the piston-rods and the air-pumps to the back ends of the rods. The "compensating cylinders" are vibrating cylinders, and are centered above and below the piston-rods, and their pistons are attached to the cross-heads of the piston-rods so that the "compensating cylinders" are vertical at half stroke and inclined at either end of the stroke. A pressure of 158 pounds of air is kept up in a small air vessel by a small air-pump, the air pressing upon water that passes in and out of the "compensating cylinders" and acts on their pistons with a continuous pressure.

The whole of the power of the steam first cut off and expanded in the high-pressure cylinder and then passed into a steam-jacketed reservoir, and from that into the low-pressure cylinder, where it is cut off and expanded much more, is developed upon the one rod, which at once takes hold of the pump-plunger, and thus the motion is perfectly direct without the intervention of any parallel motion, beam, connecting rod, crank, shaft, or fly-wheel, and consequently the friction of the engine and pump is very small, so that about 91.5

per cent. of the power of the steam is utilized, with the result that 112 pounds of coal does about 109,000,000 foot-pounds duty, or say, 2.034 pounds of coal per water horse-power, assuming the pumps to fill, which they have every chance of doing.

DIMENSIONS, ETC.

Engine.

High-pressure cylinders, - - - - -	27 in. dia.
Low " " - - - - - -	54 "
Stroke, - - - - - - - - -	3.603 ft

Pumps.

Diameter of main pumps, - - - - - -	28 in.
Stroke, - - - - - - - - -	3.603 ft.

Boilers.

Three double-flued boilers, each, - - -	7 ft. by 26 ft.
Two flues, - - - - - - - - -	2 ft. 9 in. dia.
Grates in each boiler equal, - - - -	27½ sq. ft.

Observations.

Steam pressure on trial, average, - - -	79.8 lbs.
Head of water on pumps, " - - -	148.48 ft.
Temperature injection, - - - -	46 deg.
" air-pump discharge, - - -	76.2 "
" feed-pump delivery, - - -	49.6 "
Average total strokes per minute, - - -	35.312
Vacuum, - - - - - - - - -	28 in.
Barometer, - - - - - - - -	30.2 "

Results.

Total indicated horse-power, - - - - -	330 h. p.
Pump horse-power, calculated from full displacement, - - - - - - - -	301.92 h. p.
	per cent.
Mechanical efficiency, - - - - - -	91.5
Friction, - - - - - - - - -	8.5
	100.0

Coal used per indicated horse-power per hour, including the ashes,	1.905 lbs.
Allowing for difference of temperature between cold feed and hot-well, viz., 26.5 deg.,	1.86 "
Coal used per pump horse-power (as above), including the ashes,	2.082 "
With the same deduction for temperature as above,	2.034 "
Water evaporated per pound of coal, excluding that condensed in jackets which were in circulation and not measured,	7.74 "
Water as steam used in cylinders per indicated horse-power per hour (excluding that condensed in jackets),	14.75 "
Water as steam used in cylinders per pump horse-power (as above),	16.13 "
Duty of 112 pounds of coal, including ashes,	106,500,000 ft-lb.
" allowing for difference of temperature of feed-water (as above),	109,000,000 "

I am, yours truly,

E. A. COWPER.

REPORT BY MR. E. L. MORRIS.

Results of a Coal Trial made with a High-Duty Compound-Condensing Worthington Pumping Engine erected at the New River Company's Works at Stoke Newington, London, by Messrs. James Simpson & Co., Limited, 1888.

DATE.	Hours at Work.	Strokes Made.	Strokes per Minute.	Gallons of Water Pumped.	Head of Water in Feet.	Coal Used.			Pump Horse-Power.	Pounds of Coal per P. H. P. per Hour.	Duty in Millions.
1888.						tons.	cwt.	qrs.			
Wednesday, Nov. 7th.....	8¾	8,500	16.2	3,127,000	147.0	2	4	0	264.8	2.1	105,600,000
	12	12,000	16.6	4,415,000	147.25	3	4	0	272.1	2.1	105,600,000
Thursday, Nov. 8th......	12	12,000	16.6	4,415,000	147.0	3	0	0	271.5	2.0	110,800,000
	12	12,000	16.6	4,415,000	146.0	3	0	0	269.7	2.0	110,800,000
Friday, Nov. 9th.........	12	12,400	17.2	4,562,000	147.0	3	0	0	281.2	2.0	110,800,000
	11¾	12,200	17.2	4,488,000	145.5	3	0	0	278.5	2.0	110,800,000
Saturday, Nov. 10th......	6½	7,100	18.2	2,612,000	147.25	1	8	0	....	..	....

Steam pressure in boilers - - - - 80 lb. on sq. in.
Vacuum in inches of mercury - - - - 27½
Barometer - - - - 30 in.
Temperature of injection water - - - - 46° Fahr.
Temperature of air pump discharge - - - 76° Fahr.
Cut-off high-pressure cylinders - - - - .325 of the stroke.

N. B.—Steam was supplied by three Lancashire boilers, 7 feet diameter by 26 feet long; diameter of flues, 2 feet 9 inches; length of grates, 5 feet. The coal used was Nixon's navigation Welsh. The piston speed for the horse-power is taken as being the average speed of the engine throughout the day, as shown by the counter. The pump horse-power is calculated from the full displacement of the pumps. The pressure in the delivery main was very steady throughout the trial.

New River Company's Works, Finsbury Park, London, N., November 13th, 1888.

The above are the particulars of a coal trial I made here last week with our new Worthington engine extending over a period of seventy-four hours. I think the results are very satisfactory, as no special measures were adopted in order to secure a good duty.

EDMUND L. MORRIS.

WORTHINGTON PUMPING ENGINE AT THE WEST MIDDLESEX WATER WORKS.

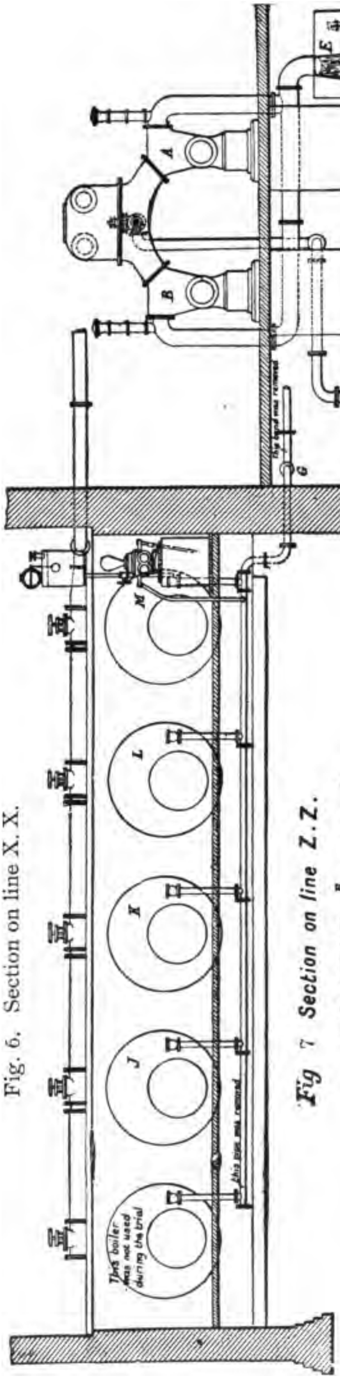


Fig. 6. Section on line X. X.

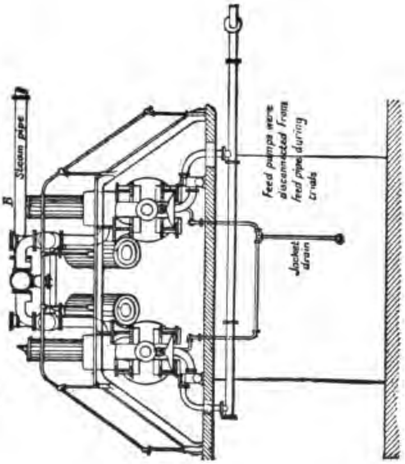


Fig 7 Section on line Z. Z.

WORTHINGTON PUMPING ENGINE AT THE WEST MIDDLESEX WATER WORKS.

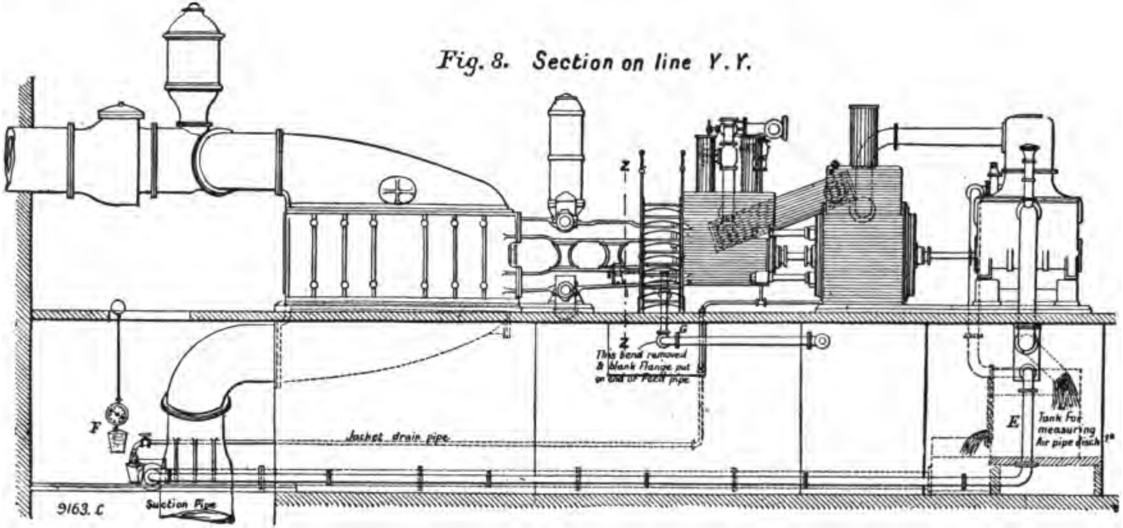


Fig. 8. Section on line Y.Y.

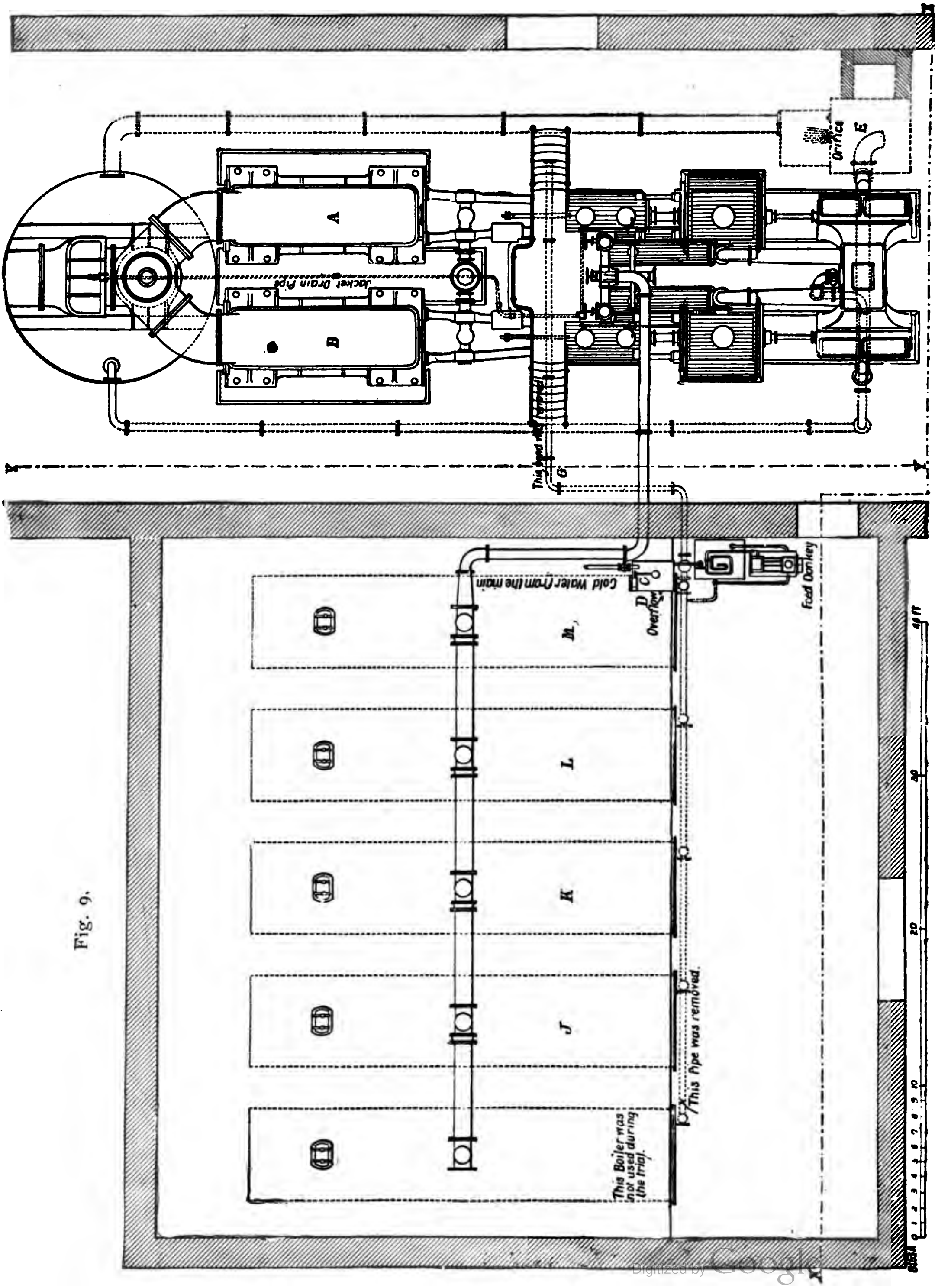

A
B
Jacket Drain Pipe
Orifice
E
This bend was removed
G
Y
Y
Cold Water from the main
C
D
Overflow
Feed Donkey
M
L
K
J
This Boiler was not used during the trial.
This Pipe was removed.
X
X
0 1 2 3 4 5 6 7 8 9 10
20
30
40 ft

Fig. 9.

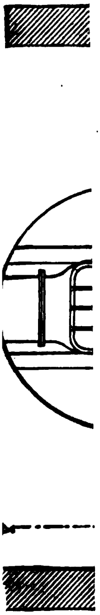


Fig. 9.

MINNEAPOLIS, MINN.

30,000,000 GALLONS CAPACITY.

TWO ENGINES.

REPORT OF PROF. WM. A. PIKE.

MINNEAPOLIS, MINN., 1890.

ANDREW RINKER, *City Engineer of Minneapolis :*

DEAR SIR—Having been requested by you to conduct the official test of the new Worthington High-Duty Pumping Engines, Nos. 433 and 434, at the North Minneapolis Pumping Station, I respectfully present the following report :

CONDITIONS OF THE CONTRACT.

The contract required that the plant should develop a "duty" of one hundred and fifteen million foot-pounds (115,000,000 foot-pounds) per one hundred (100) pounds of fuel consumed, against a total load of seventy-five (75) pounds per square inch of plunger area, and that with a piston speed not greater than one hundred and twenty-five feet (125 feet) per minute, each pump should have a capacity of fifteen million (15,000,000) U. S. gallons of water in twenty-four (24) hours. In the "duty" test it was specified that the load on the plungers should be determined by "indicator diagrams," also that to the load on the plungers, as determined by the "indicators," should be added one (1) pound for each right-angled bend in the suction pipe. In determining the capacity of the pumps it was specified that the displacement of the plungers should be considered to be the product of the area of the plungers multiplied by the total distance they traverse during the test. It was also specified that during the test steam should be furnished to the cylinders at an effective pressure of eighty-five (85) pounds per square inch above the atmosphere. By mutual agreement, before the test, in order that the pressure of the water in the city might not be too low, it was decided that the pressure against the plungers should be seventy-five (75) pounds above the atmosphere, and that steam should be used at one hundred (100) pounds above the atmosphere instead of eighty-five (85) pounds, the "duty" remaining the same.

PRINCIPAL DIMENSIONS.

The principal dimensions of the pumping engines affecting the test are as follows :

Diameter of high-pressure steam cylinder and plunger,	33.00 in.
" " low " " " " "	66.00 in.
" " main piston and plunger rods, - -	5.00 in.
" " extension plunger rods, - - - - -	3.50 in.
Net area of plunger in square inches, - - -	840.7 in.
Mean stroke of engine No. 433, - - - -	49.83 in.
" " " " No. 434, - - - -	49.645 in.

CORPS OF OBSERVERS.

The following gentlemen served very efficiently as observers, all, except Messrs. McGavin, an employee of the Worthington Company, and Redfield, of the City Engineer's office, being connected with the engineering department of the University of Minnesota :

Prof. J. H. Barr and Mr. McGavin, indicator diagrams.

M. H. Gerry, counter readings and length of stroke.

H. M. Woodward, gauge and barometer readings in engine room.

J. T. Higgins, gauge and pyrometer readings in boiler room.

T. E. Nilson, coal and water in boiler room.

J. F. Hayden, general records.

W. W. Redfield, timekeeper.

On the second day A. E. Stevens took T. E. Nilson's place and Messrs. Hoyt and Higgins changed places.

SUMMARY OF OBSERVATIONS.

Pump 434, *tested Tuesday, March* 25*th*, 1890.

Duration of test, 9 A. M. to 7 P. M., - - - - -	10 hours.
* Total coal used in ten hours, - - - - - -	9,230 lbs.
Total ash, - - - - - - - - - - -	699 "
Net fuel consumed, - - - - - -	8,531 "

---

* Lehigh Buck Mountain Vein Coal, hand picked.

Mean load per square inch on plungers as per indicator diagrams,	87 72 lbs.
Mean length of stroke,	49.645 in.
Reading of counter at 7 P. M.,	641,373
" " 9 A. M.,	632,548
Difference in ten hours,	8,825
Number of single strokes, each plunger,	17,650
Mean water gauge pressure, as corrected for error in gauge,	72.82 lbs.
Average reading of well gauge,	20.8 ft.
Mean steam pressure in boilers,	100.7 lbs.
Height of water in boiler No. 3 at 9 A. M.,	2.00 in.
" " " " 3 " 7 P. M.,	4.00 "
" " " " 4 " 9 A. M.,	4.00 "
" " " " 4 " 7 P. M.,	2.25 "

Formula for duty as specified by the contract,

$$\text{Duty} = \frac{P \times V \times H \times 100}{F}, \text{ in which}$$

P = pounds of water delivered per stroke.
V = Number of strokes during trial.
H = Total resistance to plunger in feet.
F = Number of pounds of fuel actually consumed during the trial.

$$\text{Duty} = \frac{\overset{P}{\frac{840.7 \times 49.645}{1728}} \times \overset{H}{(90.72 \times 2308)} \times \overset{V}{(35300 \times 100)}}{8531}$$

$$= \text{ - - - - - - - - - } 130{,}558{,}000 \text{ ft.-lbs.}$$

$$\text{Piston speed} = \frac{\text{length of stroke} \times V}{10 \text{ h.} \times 60 \text{ m.} \times 2} = 121.69 \text{ feet per minute.}$$

$$\text{Capacity} = \frac{A \times L \times V \times 24}{231 \times 10} = 15{,}307{,}000 \text{ U. S. gallons in twenty-four hours.}$$

If no allowance is made for bends or ashes the pressure will be 87.72 pounds, and the fuel 9230, making the "duty" 116,683,000 foot-pounds.

SUMMARY OF OBSERVATION.

Pump 433, *tested Thursday, March 27th,* 1890.

Duration of test, 9 A. M. to 7 P. M.,	10 hours.

*Total coal used in ten hours,	9,333 lbs.
Total ash,	692 lbs.
Net fuel consumed,	8,641 lbs.
Mean load per square inch on plungers as per indicator diagrams,	90.90 lbs.
Mean length of stroke,	49.83 in.
Reading of counter at 7 P. M.,	87,647
Reading of counter at 9 A. M.,	78,665
Difference in ten hours,	8,982
Number of single strokes, each plunger,	17,964
Mean water pressure as corrected for error in gauge,	77.27
Mean steam pressure in boilers per gauge,	101.7 lbs.
Average reading of well gauge,	20.78 ft.
Height of water in boiler No. 3 at 9 A. M.,	3.50 in.
" " " 3 at 7 P. M.,	6.00 "
" " " 4 at 9 A. M.,	3.50 "
" " " 4 at 7 P. M.,	3.50 "

$$\text{Duty} = \frac{P \times V \times H \times 100}{F}, \text{ as before}$$

$$= \frac{\dfrac{(840.7 \times 49.83 \times 62.4) \times (96.9 \times 2.308) \times 35928 \times 100}{1728}}{8041}$$

$$= \text{- - - - - - - - - -} \; 136{,}280{,}000 \text{ ft. lbs.}$$

$$\text{Piston speed} = \frac{\text{length of stroke} \times V}{10\text{h.} \times 60\text{m.} \times 2} = 124.32$$

$$\text{Capacity} = \frac{A \times L \times V \times 24}{231 \times 10} = 15{,}637{,}500$$ U. S. gallons in twenty-four (24) hours.

From the above results it will be seen that, strictly following the terms of the contract, both pumps develop a duty far in excess of that required, and that the capacity exceeds that guaranteed with less than the maximum piston speed.

As a matter of interest, though not strictly required by the contract, I have calculated the duty of both pumps, making no allowance for bends in the suction pipe nor for ash nor unconsumed coal, with the following results:

Pump 434 develops a duty of 116,683,000 foot-pounds per pound of coal, and pump 433 a duty of 119,179,000 foot-pounds per pound of coal. Thus both pumps develop more than the contract duty, with no allowances whatever.

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*Lehigh Buck Mountain Vein Coal, hand picked.

BOILER TRIAL.

In order to make this test more complete, I requested Prof. J. H. Barr to make an evaporative test of a pair of the boilers supplying the steam for these pumps. This test was accordingly made on Tuesday, April 9th, 1890, by the following students, under the immediate charge of Professor Barr: Messrs. Gerry, Nilson, Woodward and Aslakson.

PRINCIPAL DIMENSIONS OF BOILERS.

Inside diameter, - - - - - - - -	9 ft. 6 in.
Length between heads, - - - - - -	15 ft.
Number of 3-inch tubes, - - - - - -	118
Diameter of corrugated fire box, inside of corrugation,	39 in.
Outside of corrugation, - - - - - -	43 "
Total heating surface, each boiler, - - - -	1,557 sq. ft.
Total grate surface, each boiler, - - - -	37.5 "
Ratio of heating to grate surface, - - - -	41.5 : 1
Calorimeter or tube area, - - - - -	5.79 sq. ft.
Ratio of grate surface to calorimeter, - - -	6.48 : 1

RESULTS OF BOILER TRIAL.

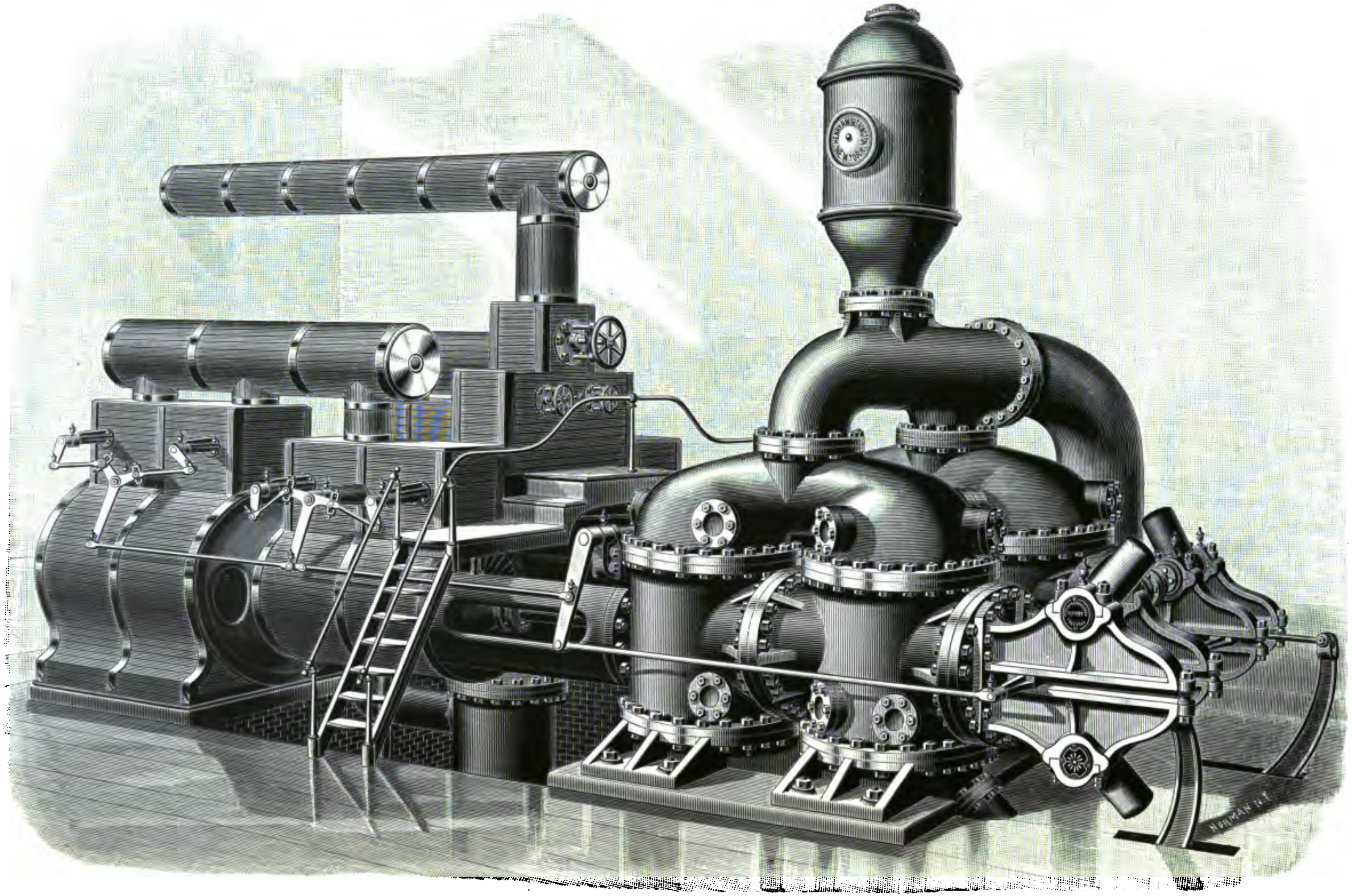
Time of test, - - - - - - - - - -	6 hours.
Mean boiler pressure above atmosphere, - - -	91.98 lbs.
Total coal used, - - - - - - - - -	5,617.0 "
Total water used as per meter, - - - - -	974.0 cu. ft.
Excess water due difference in water gauges, - -	10.83 "
Net feed water, - - - - - - - - -	963.17 "
Mean temperature of feed water, - - - - -	113.9° Fahr.
Weight of feed water, - - - - - - -	59,602 lbs.
Water per lb. of coal, as actually evaporated, - -	10.61 "
Equivalent water from and at 212° Fahr., - - -	12.10 "
Coal per hour per square foot of grate, - - -	12.48 "

It is to be regretted that a longer test than six hours could not be made, but it was desired to make the trial of the boilers with the same coal as was used during the duty test, and the supply gave out.

The meter used was tested before the trial, and no error within the limits of accuracy of the other results found.

Respectfully submitted,

WILLIAM A. PIKE.


HENRY R. WORTHINGTON NEW YORK
HERMAN N.Y.

WORTHINGTON HIGH DUTY PUMPING ENGINE AT HYDE PARK, ILL.

HYDE PARK, CHICAGO, ILL.

12,000,000 GALLONS CAPACITY.
ONE ENGINE.

REPORT OF M. L. HOLMAN, C. E.; G. H. BENZENBERG, C. E., AND CHAS. B. BRUSH, C. E.

CHICAGO, December 8th, 1890.

W. H. PURDY, *Commissioner of Public Works,*
Chicago, Ill.:

DEAR SIR—Herewith please find our report on the recent test of the new Worthington Horizontal Compound Condensing High Duty Pumping Engine at the Sixty-eighth Street Works, Chicago, Ill., with its boilers and appendages, as requested in your following letter of instructions:

CITY OF CHICAGO,
DEPARTMENT OF PUBLIC WORKS,
CHICAGO, October 18th, 1890.

MESSRS. BENZENBERG, HOLMAN & BRUSH,
Experts to Test the Worthington Engine,
Sixty-eighth Street Pumping Station.

GENTLEMEN—From the inclosed copies of proposal and specifications you will perceive that the Worthington Pumping Engine at the Sixty-eighth Street Works, with its boilers and appendages, is to be subjected to a test as to capacity, duty and strength.

The daily capacity is required to be 12,000,000 U. S. gallons when running at a piston speed not exceeding 100 feet per minute, while pumping at its capacity and duty performance.

The duty required of said engine on a 24 hour test, while pumping against a water pressure of 60 pounds to the square inch, calculated at a level with the centre of the pump plungers, shall not be less than 110,000,000 foot-pounds calculated from each 100 pounds coal actually used, and plunger displacement of pumps without any allowances, while running at a piston speed not exceeding 100 feet per minute. During such 24 hour test the steam pressure maintained at the throttle of the engine, as indicated by the gauges shall not exceed 75 pounds to the square inch. The coal used for such test shall be of the best quality, but shall be delivered without hand-picking, and the results of the 24 hour test shall be computed on the actual coal consumed under the boilers without any allowances whatever.

During such duty test speed registers shall be so placed as to show the length of each stroke, and such strokes shall average not less than 48 inches, and the water pumped shall be calculated from such registers.

The said engine shall also be subjected to a test for a sufficient duration of time to show whether there is any sign of weakness in any of its parts while pumping against a water pressure of 150 pounds to the square inch.

The boilers are to carry safely a working pressure of from 85 to 90 pounds per square inch, and must have been tested and made thoroughly tight at a cold water pressure of 125 pounds per square inch before shipment from the place of manufacture.

Your committee will make a report in writing, particularly specifying the result of the test made.

Your committee will also investigate, and report in writing, as to whether the design of the machinery and boilers is in accordance with the proposal and specifications, copies of which are inclosed herewith, and whether the material and workmanship are of the best of their several kinds, and whether the various parts are of ample strength and size, and whether the machinery and boilers are first-class in every respect as to finish and modern requirements.

Respectfully,
W. H. PURDY,
Commissioner of Public Works.

Agreeable to your instructions we met at your office on the morning of Tuesday, October 21st, 1890, and after consultation with yourself, His Honor, Mayor Cregier, and Acting City Engineer Cheney, we proceeded to the pumping station to make the necessary preparations for proceeding with the test the next day.

We found the Worthington engine at work on regular direct pressure supply in connection with the other engines at that station, and that it had been in such regular operation for several months.

After examining the plant in its various parts we again met next on Wednesday, and in connection with Messrs. Feind, Green and others, from the City Engineer's office, took the necessary measurements, levels, etc. We were unable to proceed with the test, however, until 1.30 P. M. on Thursday, October 23d, owing to delay on the part of the city in furnishing the coal agreed upon by the contracting parties.

In oollecting the necessary data we obtained the following:

DIMENSIONS OF ENGINE AND PUMPS.

The principal dimensions of the engine and pumps are as follows:

Diameter of high pressure cylinders,	33	in.
Diameter of low pressure cylinders,	66	“
Diameter of pump plungers,	33	“
Stroke of steam pistons and pump-plungers (contract),	48	“
Stroke of steam pistons and pump-plungers (actual),	49.86	“
Diameter of high pressure piston-rods,	5.5	“
Diameter of low pressure piston-rods,	5.25	“
Diameter of pump-rods from steam cylinders,	5.5	“
Diameter of pump-rods to compensating cylinders,	3.75	“

BOILERS AND CHIMNEYS.

Number of boilers,	4
Diameter of shell,	66 in.
Length of shell,	18 ft.
Thickness of shell,	0.375 in.
Thickness of head,	0.5 “
Domes,	2′ 9″ x 3′ 8″
Number of tubes,	68
Diameter of tubes,	4 in.
Length of tubes,	18 ft.
Width of fire-grate,	66.7 in.
Length of fire-grate,	54 “
Area of fire-grate (each boiler),	25 sq. ft.
Height of chimney,	100 ft.
Diameter of chimney flue,	5 ft. 6 in.

From the foregoing dimensions the following deductions are made as to proportions for each boiler:

Total area of chimney flues,	23.75 sq. ft.
Total area of grate surface,	25 “
Total heating surface, one-half area of shell plus one head plus tubes,	1,461.5 “

Total area of cross section of flues,	5.93 sq. ft.
Ratio of heating surface to grate surface,	58.44
Ratio of grate surface to area through tubes,	4.22
Ratio of grate surface to area of chimney flue	1.05

The effective area of each plunger has been determined as follows:

Area of plunger,	33 in. diam.		855.30 sq. in.
Area of pump-rod from steam Cyl.	5½ "	23.76	
Area of pump-rod to Comp. Cyl.,	3¾ "	11.04	
		34.80	
Mean,			17.40 "
Effective area of each plunger,			837.90 "

The steam cylinders are jacketed both on the sides and ends. A separator is located in the steam pipe immediately above the main throttle and in the exhaust passages between the high and low pressure cylinders are placed a series of reheating tubes. The water collected from the steam by the separator as well as that which is condensed in the reheating tubes and jackets, is returned directly to the boilers by means of two steam loops.

The feed water is taken from the delivery of the air pumps, passing into a hot-well, from which the feed pump supplying the boilers draws its supply. The exhaust of the feed pump is returned into the hot-well. The exhaust steam from the low pressure cylinders enters a jet condenser, from which the water and vapor is withdrawn by means of two air pumps respectively operated on each side of the engine, being attached directly thereto by means of links and beams, and having the same stroke as the engine itself.

The clearance space in all of the cylinders is very small.

The main steam valves of both the high and low pressure cylinders are of the Corliss rotary type and are operated in the manner usual to the duplex form of engine. The cut-off valves are also of the Corliss type and are operated respectively by each side of the engine to which they belong, being connected therewith by a positive attachment which can be altered at will to change the point of cut-off.

The steam is admitted to and exhausted from the cylinders by the main steam-valves above described, the ratio of expansion being controlled by the cut-off valves.

The pumps, which are double-acting, are fitted with rubber-disc valves of the usual Worthington type, each valve being 5 inches in diameter and having ⅝ inches lift.

At each end of each pump there are 36 induction and 36 eduction valves, giving an ample valve opening for the reception and discharge of the water.

The products of combustion pass from the furnaces back under the boilers forward through the tubes, and back over the top to the chimney in connection with the rear ends.

The following elevations are based on the Chicago datum—*i. e.*, low-water mark in Lake Michigan, A. D., 1847. The lake at times has been 0.70 feet below and 4 feet above datum.

Surface of water in Lake Michigan, - - - -	+ 0.93 ft.
Gauge in open well, - - - - -	+ 9.00 "
Top of well, - - - - - -	+ 7.60 "
Bottom of well, - - - - -	— 37.66 "
Foot valve on suction, - - - -	— 23.80 "
Water in well, - - - - - - -	— 0.45 "
Engine-room floor, - - - - -	+ 8.13 "
Tap for vacuum gauge on suction, - -	+ 8.42 "
Zero on " " " - -	+ 11.65 "
Centre of plunger and lower water-gauge,	+ 12.29 "
Upper water-gauge, - - - - -	+ 20.67 "
Steam-gauge, - - - - - -	+ 22.00 "
Tap for steam-gauge, - - - -	+ 24.30 "
Vacuum-gauge on condenser, - - -	+ 22.01 "
Bottom of boilers, - - - - -	+ 11.44 "
Top of tubes in boilers, - - - -	+ 14.85 "
Zero on water-gauges of boilers, - -	+ 14.85 "

The well from which the water is taken by suction for the Worthington Engine is located outside of the building and is about 10 feet in diameter. The well is connected with a tunnel 5 feet in diameter and about 300 feet in length to the edge of the lake. From this point the tunnel runs out under the lake 6 feet in diameter for a distance of about one mile.

The coal agreed upon by and between yourself and the representatives of the contractor having been delivered during the forenoon of October 23d, it was determined to commence the test at one o'clock on that day and continue the same until one o'clock Friday noon. It was further determined what records and observations should be collected during the trial run. The engine was doing its part of the work in supplying the general distribution. It having been determined that Boilers Nos. 1 and 2 only were to be used in supplying steam to the engine, the gates in the steam-pipe, connecting these boilers with the other boilers in the building, were closed, and a 2-inch tap made into the steam-pipe leading from the old battery of boilers, which was left open during the test. All necessary preparations, as the testing of all gauges, scales, &c., of cleaning the ash pits, floors, the attaching the various instruments for taking the observations and of noting the condition of the fires, height of water, &c., having been made, the test was commenced at 1.03 P. M. Thursday, and the following data at each 15 minutes interval were collected.

No corrections of any kind have been made in the following table of general and average results:

Amount of coal weighed and delivered into the furnaces during the 24 hour test,	14,654 lbs.
Amount of ashes obtained from coal used,	882 “
Difference of readings of feed-water meter,	2,032 cu. ft.
Temperature of feed-water, average,	135.95° Fahr.
Temperature of air in boiler room, average,	74.43° “
Temperature of steam in boilers, average,	321.5° “
Temperature of gasses in flue, average,	404.5° “
Difference in pressure of atmosphere between inside and outside of flue, draft gauge,	.4 in.
Steam pressure at boiler No. 1, average,	73.57 lbs.
Steam pressure at boiler No. 2, average,	75.80 “
Steam pressure at boiler No. 3, average,	58.79 “
Steam pressure at boiler No. 4, average,	56.91 “
Height of water in boiler No. 1 above top of tubes, average,	4.08 in.
Height of water in boiler No. 2 above top of tubes, average,	4.38 “
Height of water in boiler No. 3, above top of tubes, average,	11.99 “
Height of water in boiler No. 4, above top of tubes, average,	8.25 “
Deduction from contact stroke No. 1, side of engine, average,	0.233 “
Deduction from contact stroke No. 2, side of engine, average,	0.221 “
Double contact stroke, water cylinder,	8.31 ft.
Length of observed double stroke, average,	8.272 “
Average number of double strokes per minute,	12.112 “
Height of barometer, average,	29.502 in.
Temperature of engine room, average,	77.14° Fahr.
Temperature of water in open well, average,	56.00° “

Weight of water at temperature 56° Fahr., - -	62.35 lbs.
Elevation of water in open well, average, - -	—0.034 ft.
Average height from level of water in suction well to center of plungers, - - - - - -	12.324 "
Average height from level of water in suction well to center of plungers, - - - - - -	5.335 lbs.
Temperature of water in air pump delivery, average,	91.78° Fahr.
Temperature of outside air, average, - - -	48.73° "
Number of double strokes of each pump, - -	17,441
Pressure on upper water gauge, average, - -	55.665 lbs.
Pressure on steam gauge at engine, - - -	75.51 "
Vacuum on condenser, - - - - - - -	26.98 in.
Vacuum on suction, average, - - - - -	8.6 "
Water gauge on level with centre of plungers, average,	61.83 lbs.
Weight of one cubic foot of water as measured by meter at 110° Fahr., - - - - - - -	62.475 "
Weight of one cubic foot of water at 110° Fahr., -	61.8296 "
Weight of one cubic foot of water at 135.95° Fahr,	61.42 "
Ratio of weight of one cubic foot of metered water at 110° Fahr. to correct weight, - - - - -	1.043 per cent.

After the completion of the test the various gauges and scales were retested, and the feed-water passing through meter accurately weighed, from which the following corrections were deduced:

On the water gauge at the level of the centre of pump plungers we have subtracted .375 pounds from each reading.

On the steam gauge at the engine we have added 0.50 pounds to each reading for correction of gauge and subtracted one pound for level of tap.

On the steam gauges at the boilers:

At boiler No. 1 we added 2.25 pounds.

At boiler No. 3 we added 2.5 pounds.

At boiler No. 4 we subtracted 1.5 pounds.

We have corrected the meter readings by adding 1.043 per cent. to each reading. We have subtracted 0.038 feet from 8.31 feet, double contact stroke in the water cylinders, making the length of the observed double stroke 8.272 feet.

We have estimated the jacket water at 10 per cent. of the water measured by the meter, and have corrected for the difference of level of water in boilers.

We have also added to the average pressure of the gauge at the centre of the plungers the average distance from the level of suction well to the centre of the plungers in ascertaining the duty on the basis of coal used.

The scales were tested and sealed before the test began by the proper city authority, but at the expiration of the test the scales, having been again tested, were found to indicate uniformly one pound less than the actual weight at each weighing. We have therefore added one-half pound for each weighing of coal, thus making a total correction of 29.5 pounds, and making the total amount of coal weighed and delivered into the furnaces 14,683.5 pounds.

The conditions of the fires were not as good at the end as at the beginning of the test. Having made careful observations of the condition of the fires at both ends of the test, we have computed the additional amount of coal necessary to put the fires at the end of the test run in good condition to be 203 pounds, and have, therefore, added this amount to the corrected amount of coal weighed and delivered into the furnaces as the actual coal consumed under the boilers during the 24 hours test. This makes total amount of coal to be used in computing the duty of 14,886.5 pounds. With corrections, we arrive at the following results:

Actual coal consumed under the boilers, - -	14,886.5	lbs.
Water gauge on level with centre of plungers, -	61.455	"
Pressure on steam gauge at engine, - - -	75.01	"
Corrected weight of one cubic foot of metered water at 135.95° Fahr., - - - -	62.06	"
Total amount of metered feed water, - -	126,106.	"
Total amount of feed water including 10 per cent. for jacket water, - - - - - -	138,717.	"
Average amount of feed water per minute, -	96.33	"
Average amount of coal used per minute, - -	10.34	"
Average pounds of feed water evaporated per pound of coal, - - - - - -	9.32	"
Average pounds of coal per square foot of grate per hour, - - - - - - -	12.41	"

We also find the

Average horse-power of engine based on water column to be

$$\frac{837.9 \times 66.79 \times 100.19 \times 2}{33,000} = 339.82 \text{ horse-power.}$$

Coal per above horse-power per hour, - -	1.83	"
Water evaporated per horse-power per hour, -	17.00	"

The capacity of each pump based upon the plunger displacement is as follows:

$$\frac{837.9 \times 49.632 \times 17,441 \times 2}{231} = 6,279,765 \text{ U. S. gallons.}$$

Or for both pumps, 12,559,530 U. S. gallons, which exceeds the specified capacity 4.65 per cent.

We arrive at the duty developed by the engine in doing the work during the 24 hours of the trial run by the following formula:

$$\frac{837.9 \times 8.272 \times 17,441 \times 2 \times 66.79}{14,886.5} \quad 100$$

$$=108,473,320 \text{ foot-pounds.}$$

We have taken a number of cards during the test, all of which show a fairly satisfactory distribution of steam. There is a little wire drawing in the admission line, which, however, is not serious, and the rounding of the line at the point of cut-off might be criticized, but the loss there is trifling and is compensated for in the simplicity of the cut-off valve. The expansion line is excellent, following very closely the theoretical Isothermal line. The expansion line in the low pressure cylinder, after the point of cut-off, follows very closely the theoretical, and the friction between the two cylinders, high and low, as indicated by the cards taken on the same sheet of paper, is very small. From the low pressure terminal the steam passes into the condenser with but a small amount of friction in the passages and intermediate pipes.

We believe this engine can be safely operated at a piston speed of 125 feet per minute, which is equivalent to a capacity of 15,000,000 gallons in 24 hours. The valve areas, steam passages and other parts of the engine are amply sufficient to enable 15,000,000 gallons to be pumped daily with the steam pressure allowed in the contract. During the test the full steam pressure permitted by the contract was not used, as can be seen by an examination of the high pressure cards, the main throttle having been partly closed in order to keep the engine within the condition prescribed.

We submit herewith a number of indicator cards taken during the test at the points and at the times as shown on each card. We also submit a copy of all the observations taken during the trial run and also of the tests made of the gauges used, with the exception of the steam gauge on boiler No. 2, which could not be taken down.

Having now described the engine and its boilers and appendages, the conditions under which the tests were made and the results obtained, based on the usual methods of testing pumping engines, we will next consider the specific requirements of the contract, as contained in your letter of instructions.

First.—" The daily capacity is required to be 12,000,000 U. S. " gallons when running at a piston speed not exceeding 100 feet a " minute, while pumping at its capacity and duty performance."

The capacity of the engine based upon the plunger displacement is 12,559,530 U. S, gallons in 24 hours, while running at an average piston speed of 100.19 feet per minute and while pumping at its capacity and duty performance. This exceeds the specified capacity by 4.65 per cent. when running at 100 feet per minute.

Second.—" The duty required of said engine on a twenty-four " hour test, while pumping against a water-pressure of 60 pounds to " the square inch, calculated at a level with the centre of the pump " plungers, shall not be less than one hundred and ten millions foot- " pounds, calculated from each 100 pounds coal actually used, and " plunger displacement of pumps without any allowances, while run- " ning at a piston speed not exceeding 100 feet per minute. During " such twenty-four hour test the steam-pressure maintained at the " throttle of the engine, as indicated by the gauges, shall not exceed " 75 pounds to the square inch. The coal used for such test shall be " of the best quality, but shall be delivered without hand-picking, " and the results of the twenty-four hour test shall be computed on " the actual coal consumed under the boilers without any allowances " whatever.

" During such duty test-speed registers shall be so placed as " to show the length of each stroke, and such strokes shall average " not less than 48 inches, and the water pumped shall be calculated " from such registers."

We found the duty of the engine during the twenty-four hour test, while pumping against a water-pressure of 61,455 pounds to the square inch, calculated at a level with the centre of the pump plungers to be 108,473,320 foot-pounds, calculated for each 100 pounds of coal actually used, and plunger displacement without any allowances, while running at an average piston speed of 100.19 feet per minute. During said twenty-four hour test the average steam-pressure maintained at the throttle of the engines was 75.01 pounds per square inch. The coal used was purchased by you as best quality Cross Creek Anthracite coal, not hand-picked. It was delivered in a closed car, No. 8414, Union Line, Pa. Co., and put by your men into the coal-shed at your Sixty-eighth Street Works, and delivered to the contractor as received. The results of the twenty-four hour test above given, were computed on the actual coal consumed under the boilers during the twenty-four hour test, without any allowances whatever. The length of the stroke of each pump was determined by placing scales on each side of the engine and pointers placed on each pump rod, and was found for the entire run to be 49.632 inches.

Third.—"The said engine shall also be subjected to a test for "a sufficient duration of time to show whether there is any sign of "weakness in any of its parts while pumping against a water pressure of 150 pounds to the square inch."

This test was made by pumping against a closed gate, a 6-inch by-pass gate being left sufficiently open from the delivery chamber on the top of the pump to enable the engine to run up to the required pressure. This rather hazardous operation was rendered necessary in consequence of the objection made against subjecting the city mains to a pressure of 150 pounds per square inch.

About an hour was occupied in working the pumps up from a pressure, at the centre of the plungers, of 60 pounds to 150 pounds per square inch. During 40 minutes the pump worked under a pressure ranging from 100 to 140 pounds, and during 10 minutes from 140 to 150 pounds; the gauge used in this test was subsequently tested and was practically correct.

Fourth.—"The boilers are to carry safely a working pressure "of from 85 to 90 pounds per square inch, and must have been "tested and made thoroughly tight at a cold water pressure of 125 "pounds per square inch before shipment from the place of manufacture."

Since the test of October 23d and 24th two of the boilers have been subjected in our presence to a working pressure of 90 pounds per square inch without injury to the same. As to the cold water test, we have no personal knowledge of what tests were made before shipments from the place of manufacture.

In conclusion, we submit that in our opinion the design of the machinery and boilers is in accordance with the proposals and specifications—that the material and workmanship are the best of their several kinds—that the various parts are of ample strength and size, and that the machinery and boilers are first class in every respect.

Recognizing the courtesy, constant attention and ready assistance given us by yourself and the employees of the City Engineer's Department while engaged in this test,

We have the honor to be, Respectfully yours,

M. L. HOLMAN,
G. H. BENZENBERG.

I agree with the above report except to the duty as determined, for reasons stated in my supplementary report herewith attached.

CHAS. B. BRUSH.

SUPPLEMENTARY REPORT OF CHAS. B. BRUSH, C. E.

CHICAGO, ILL., December 8th, 1890.

W. H. PURDY, ESQ.,
Commissioner of Public Works, Chicago, Ill.

DEAR SIR—I agree with all that my associates, Messrs. Benzenberg and Holman, have stated in their accompanying report to you of this date in relation to the tests of the Worthington engine, boilers and appendages at the Sixty-eighth Street Works, Chicago, Ill, made October 23d and 24th, 1890, except as to the propriety, under the terms of this contract, of adding to the total amount of coal weighed and delivered into the furnaces the results of the computation of the difference in the condition of the fire at the beginning and at the end of the test. I agree with them that the fire was not as good at the finish as it was at the start. I also agree with them in the computation that the difference in the condition of the fire amounted to 203 pounds of coal.

This contract specified how the test is to be conducted. The piston speed allowed is slow, and yet this speed need not, under the terms, be maintained. The steam pressure cannot exceed 75 pounds, but can be as much lower as will suffice to run the engine.

The computation of the duty is to be made on the coal used without any allowances whatever. Often, when an engine is required by the terms of the contract to show the duty on coal, an allowance is made of the amount of coal necessary to make the fires fresh at the start and to bring them up to the same condition at the finish.

For instance, the contract for the Worthington engine at Memphis, made in 1888, just previous to the contract for the Chicago engine, provides, "All the coal used during the trial shall be charged to the boilers, without deduction of any kind; at the close of the trial the fire must be in as good condition and the water in the boilers at the same height, and the steam at the same pressure as at the beginning of the trial."

But in your test it was evidently not considered necessary to charge any coal except that actually fed to the boilers during the twenty-four hour run, because the provision was inserted that no allowances whatever were to be made. To insist that the fires should be brought to the same point at the finish as they were at the start is to bring into the record an element of estimate and calculation that it was considered desirable to eliminate. The engine could properly have been run so long as steam was left in the boilers to drive it. The contract merely specifies that it shall not be run at a speed greater than one hundred feet per minute. It does not state that it shall not be run at a lower speed than this, and no restriction whatever is put upon allowing the fires to burn down to a point where they would merely maintain the engine motion, provided this is done within the limit of the time prescribed for the test.

I believe that under the terms of this contract the total amount of coal to be used in computing the duty should be the coal weighed and delivered into the furnaces during the twenty-four hour run—viz., 14,683.5 pounds.

I therefore ascertain the duty on coal without allowances to be as follows:

$$\frac{837.9 \times 8.272 \times 17{,}441 \times 2 \times 66.79 \times 100.}{14{,}683.5.}$$

$$= 109{,}973{,}000 \text{ foot-pounds.}$$

If the result of a computation as to the difference of fires (203 pounds) is to be added to the coal weighed and delivered into the furnaces, a correction should be made by subtracting that portion of the coal which was not consumed—viz., 882 pounds of ashes.

This computation and correction would give a duty on coal of

$$\frac{837.9 \times 8.272 \times 17{,}441 \times 2 \times 66.79 \times 100}{14{,}004.5}$$

$$= 115{,}305{,}000 \text{ foot-pounds.}$$

I believe, however, that both this computation and correction are really allowances, and under the terms of the contract neither should be made.

During this test we obtained results in detail as to the working of the plant which, in my opinion, are so important as to make it desirable that they should be permanently recorded. I therefore particularly specify them in this report.

I have included in all that follows in this report and in all that is shown on my exhibits, the 203 pounds of coal computed for the difference of fires, and do so because by so doing I agree with all the data shown in our joint report, and because, waiving the limitations of the contract, I admit that it is the proper manner of determining the amount of coal used. In the following pages, therefore, my only assumption is in distributing uniformly these 203 pounds of coal throughout the test. The method of distribution will make no practical difference in arriving at the result, and the method I have adopted is the most convenient, and, perhaps, the least open to criticism.

Results obtained after running 3, 6, 9, 12, 15, 18, 21, *and* 24 *hours, making the corrections as above stated, and taking the average or total for each cumulative period:*

TIME.	Average Length of Double Stroke at End of Each Period.	Total Number of Double Strokes of Each Pump at End of Each Period.	Average Load in Pounds per Sq. In. at End of Each Period.	Total Pounds of Water Evaporated at End of Each Period.	Total Pounds of Coal Used at End of Each Period, Including Computation for Difference in Fires.	Effective Area of Plungers in Sq. In.
October 23d, 1890.	Feet.		Lbs.	Lbs.	Lbs.	
4.03 P. M........	8.265	2,250	70.319	17,466	1,729	837.9
7.03 "	8.267	4,466	68.354	34,801	3,357	837.9
10.03 "	8.272	6,601	67.436	51,696	5,086	837.9
October 24th.						
1.03 A. M........	8.275	8,717	67.231	68,853	7,014	837.9
4.03 "	8.273	10,893	66.987	85,905	9,144	837.9
7.03 "	8.274	13,015	66.871	102,573	10,972	837.9
10.03 "	8.272	15,245	66.792	120,302	13,001	837.9
1.03 P. M........	8.272	17,441	66.790	138,717	14,886.5	837.9

From data given in above table the profiles and charts were prepared, which appear on annexed Exhibits No. 1 and No. 2.

EXHIBIT NO. 1.

The upper profile shows the evaporation per pound of coal delivered into the furnaces and the amount computed for difference in fires, calculated at the end of each three-hour period. A good evaporation was maintained until the end of the sixth hour, from which time it fell gradually until the fifteenth hour, and from then on it remained practically constant to the termination of the test.

The next profile shows the duty of the engine, based both upon the coal delivered into the furnaces and the amount computed for the difference in fires, and upon the coal per assumed evaporation of 10 pounds of water per 1 pound of coal, calculated at the end of each three-hour period. These results, which are brought out more clearly, perhaps, by Exhibit No. 2, indicate that the duty based upon the feed water was nearly uniform throughout the test, while that based upon coal was high during the first six hours, and then dropped off quite rapidly until the fifteenth hour, remaining nearly constant until the end of the test. This indicates that the economy of the engine itself was practically constant, but that the economy of the entire plant, both of engines and boilers, was affected by the evaporation, as above described.

The revolutions per each fifteen minutes show quite a range in variation, but when taken in connection with the lines of the water pressure, indicate the ability of the engine to maintain a uniform pressure while operating under the wide ranges of speed due to direct service.

The line of the steam pressure at the engine, as shown, is not particularly uniform.

The line of the water pressure is shown as indicated by the pressure gauge, located at the centre of the plungers.

EXHIBIT NO. 2.

The black line represents foot-pounds. This shows that the work performed by the engine was very nearly, if not quite, constant during the entire run of 24 hours. The engine maintained an average piston speed of 100.19 feet per minute, and an average water pressure of 61.455 pounds per square inch, on level with centre of plungers.

The red line represents the amount of coal delivered into the furnaces and the amount computed for difference in fires, and when taken in connection with the black line, it shows that the efficiency of the plant was greatest during the early portion of the run, falling, quite materially, at about the middle of the test, after which time it rises slightly, and then remains about uniform until the end of the trial.

The blue line indicates the pounds of water evaporated.

When this blue line is taken in connection with the red line it is apparent that the falling off in the duty during the latter part of the run was due to the decreasing rate of the evaporation of the boilers.

The amount of water evaporated during the twenty-four hour run, after making corrections for the meter, gauges, height of water in boilers, allowing 10 per cent. for jacket water, and taking the weight of a cubic foot of water at the different temperatures as given in Nystrom's Mechanic's, was 138,717 pounds.

The duty obtained on the evaporation of this amount of water, based upon the usually assumed evaporation of 10 pounds of water to 1 pound of coal, was

$$\frac{837.9 \times 8.272 \times 17{,}441 \times 2 \times 66.79 \times 100}{138{,}717 \div 10}.$$

$$= 116{,}409{,}000 \text{ foot-pounds.}$$

From thirteen indicator cards taken on the high and low-pressure cylinders successively, at different times, ranging from 2.30 P. M. October 23d, to 4 A. M. October 24th, I have obtained the following average results:

Mean effective pressure in the high-pressure cylinder,	32.32	lbs.
Mean effective pressure in the low-pressure cylinder,	10.015	"
Pressure, steam terminal, low-pressure cylinder,	6.67	"
Well gauge, mercury,	8.488	in.
Well gauge,	4.15	lbs.
Distance from well gauge to pressure gauge on level with centre of pump plungers,	1.67	"
Pressure on water gauge at level with centre of pump plungers,	60.687	"
Total head,	66.507	"
Total head reduced to low-pressure cylinder,	16.626	"
Total mean effective pressure, steam end, reduced to low-pressure cylinder,	18.096	"

From the above data it appears that the efficiency of the engine amounts to 91.88 per cent.

Respectfully submitted,
CHAS. B. BRUSH.

SECOND TEST.

REPORT OF J. J. DE KINDER, C. E.; T. T. JOHNSTON, C. E., AND JOHN LUNDIE, C. E.

Chicago, Ill., March 12th, 1891.

A. W. Cooke, Esq., *City Engineer.*

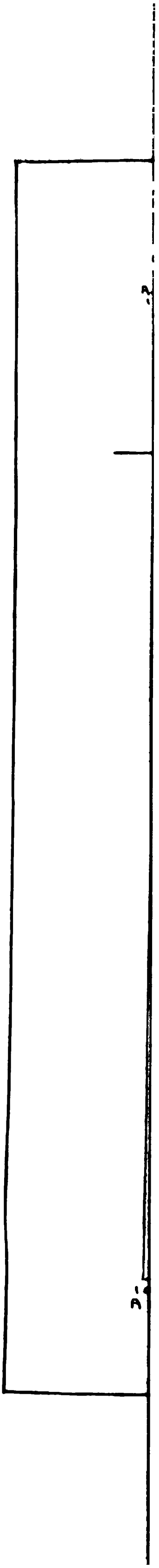
Dear Sir—Your Committee of Experts, appointed to make a test of the Worthington pumping engine at the Sixty-eighth Street Water-Works, submit the following report;

Observations were commenced at 3 p. m., Tuesday, March 10th, and ended at 3 p. m. of the following day.

All communications between the boilers in use during the test and the remaining boilers at the station were closed and provisions made to prevent any possible leakage past the communicating valves.

The condition of the fires at the commencement of the test was accurately observed, and the fires were left in the same condition at the end thereof. The water level in the boiler was observed to be 3½ inches at the beginning, and 4¼ inches at the end of the test.

All necessary levels were accurately established prior to the commencement of operations.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DUTY.

The duty developed during the aforesaid twenty-four hours was 118,043,000 foot-pounds, calculated on a basis of 100 pounds of coal consumed (as per your letter of instructions).

The following is a summary of the various observations made during the test:

Average steam pressure at engine - - - -	74.9 lbs. per sq. in.
Average water pressure at gauge - - - -	61.45 lbs. per sq. in.
Average total head pumped against - -	164.27 ft.
Average length of stroke - - - -	49.72 in. = 4.143 ft.
Average piston speed per minute - - -	100.4 ft.
Total number of strokes - - -	69,776
Temperature of water in well - - -	37° Fahr.
Number of lbs. of coal charged into furnaces -	14,612.
Number of gallons water pumped - - -	12,582,600

Respectfully submitted.

J. J. DE KINDER,
T. T. JOHNSTON,
JOHN LUNDIE.

THIRD TEST.

REPORT OF JOHN LUNDIE, CIVIL ENGINEER.

CHICAGO, March 27th, 1891.

A. W. COOKE, ESQ., *City Engineer:*

DEAR SIR—I beg to report result of test of the Worthington Pumping Engine at the Sixty-eighth Street Water-Works made on 23d inst.

The test commenced at 10.45 A. M. and finished at 8.45 P. M., being of 10 hours duration.

The boilers used were the same as were used during the test of March 10th and 11th, made by Messrs. De Kinder, Johnston and myself. All communications between the boilers used in the test and the remaining boilers at the station were shut off.

The conditions of the fires at the end of the test were, in my judgment, practically the same as at the beginning. However, to prevent any doubt in the matter of coal consumption, I had a careful record kept of the time of charging furnaces, together with the number of shovelfulls * of coal charged, so that I am enabled to submit herewith a diagram showing the rate of coal consumption, from which useful results may be deduced other than concern this test.

I placed the standard test gauge on the water column so that this, the most important gauge reading, might be unquestioned.

The duty developed during the test was 113,430,000 foot-pounds per 100 pounds of coal calculated on the basis of plunger displacement.

The amount of water pumped during the 10 hours aforesaid was 5,203,435 gallons plunger displacement, which is equivalent to 12,488,244 gallons per 24 hours.

The piston speed developed averaged 99.62 feet per minute.

The comparative work indicated by diagrams from steam and water cylinders is being worked up, and will be presented to you at an early date.

SUMMARY OF OBSERVATIONS.

Average steam pressure as indicated by gauge at throttle,	77.02 lbs.
Average water pressure as indicated by gauge on water column,	62.24 "
Elevation of water gauge above datum,	20.75 ft.
Average elevation of water in well,	—0.76 "
The above gives head pumped against, of,	165.08 "
Average length of stroke,	49.64 in.
Number of revolutions,	7,225
Total coal charged into furnaces,	6,319 lbs.

Respectfully submitted,

JOHN LUNDIE,

Civil Engineer.

APPENDIX.

DATA USED IN CALCULATIONS.

Mean area low pressure piston,	3410.37 sq. in.
" " high " "	832.60 "
" " plungers,	837.90 "
" " piston speed,	99.62 ft. per min.

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* The weight of this unit was determined by dividing the weight of the total amount of coal used by the number of shovelfulls charged to furnaces.

HORSE-POWER DEDUCED FROM INDICATOR DIAGRAMS.

		Steam.	Water.
South Engine.	I. H. P. low pressure cylinder, -	123.851	
	“ high “ “ -	87.518	
	“ water “ “ -		191.778
North Engine.	“ low pressure cylinder, -	119.527	
	“ high “ “ -	91.112	
	“ water “ “ -		195.349
	Total indicated horse-power, -	422.008	387.127

Ratio of indicated horse-power of water end to
that of steam end, - - - - - - 91.72 per cent.

Duty calculated on indicated horse-power of water end:
121,303,000 foot-pounds per 100 pounds of coal.

Respectfully submitted,

JOHN LUNDIE,
Engineer in Charge of Test.

CHICAGO, March 31st, 1891.

JOHN LUNDIE,
B. Sc. M. Am. Soc. C. E.
CIVIL ENGINEER.

1105 THE ROOKERY,
CHICAGO, July 21st, '91.

HENRY R. WORTHINGTON,
95 Lake Street, City.

DEAR SIR—Your favor of the 17th inst. received, and in reply to your inquiries will say that the test of the Hyde Park Pumping Engines, made on the 23d of March, 1891, was under my supervision, at the request of the city authorities.

The engine had no special preparation, and the boilers were fired by the firemen at the station, none of whom had any special knowledge in regard to the proper method of handling hard coal.

The coal used was ordinary merchantable coal, bought in the open market of Chicago.

Very truly yours,

JOHN LUNDIE.

OXFORD, ENGLAND.

4,250,000 GALLONS CAPACITY.

ONE ENGINE.

REPORT OF W. H. WHITE, C. E.*

We have on several occasions illustrated the Worthington pumping engine; and as this engine is now becoming very largely used in England, and is almost entirely replacing the old-fashioned type of beam engine—and also all other types of fly-wheel engines—the trials and data now given will be of much interest to our readers. One of the greatest proofs of the perfect manner in which these engines work is shown by the fact that a large number of water works companies who have them already at work are putting down duplicate or other sizes of Worthington engines. For instance, the West Middlesex Water Works Company commenced by putting down a small engine for sand washing at Barnes, and so pleased were they with it that they put down an engine of about 250 horse-power for forcing the water from the intake at Hampton to the reservoirs and filter beds at Barnes.

This engine has for the last three years been doing the same services as previously were done by Bull engines, and by its use the saving to the West Middlesex Water Works Company has been about £800 per year, and the following were the remarks of Sir W. H. Wyatt, the chairman, at one of the annual meetings of the company: "Two of the old Cornish Bull engines at Hammersmith were being replaced by one Worthington engine, which would be of greater horse-power than the other two together. The new Worthington engine at Hampton had saved them last year more than 800 tons of coal, which represented not only a large money saving, but was of great importance in these days of strikes, when there was so much uncertainty about their coal supplies." Since that time the Worthington engine at Hammersmith has been started and set to work, and is also effecting a very large saving in fuel over pumping the same quantity of water by means of the Cornish engines previously in use there. Again, at Sydney, N. S. W., a large Worthington engine was sent out under the advice and inspection of Sir John Fowler, Bart., K. C. M. G., and his decision on the type of engine has been justified in the most satisfactory manner, by an order having been sent over from Sydney for a duplicate, so pleased were the authorities at Sydney with the working of the first engine. The illustration given is of an engine that is at work at the Oxford Water Works, and of two similar ones which are at work in Berlin, and when Mr. E. K. Burstal, M. Inst. C. E., Westminster, who was then engineer to the Oxford Water Works, decided on adopting a Worthington engine, the Corporation, before agreeing to adopt that type, obtained an opinion from one of the most distinguished engineers, who is a recognized authority on pumping engines—viz., Mr. E. A. Cowper, M. I. C. E., and the following is an extract from his report, as printed in the Oxford *Times:* "I have had the advantage of learning a number of facts in reference to the quantity of water that you require to pump, and the probable future increase, etc., from Mr. Burstal, your engineer, and I find, on constructing a diagram of the increase in population, that in seven years the present number of about 50,000 will become 60,000, and, therefore, you should, of course, be prepared to pump an increased quantity when required. Now, if your present consumption of twenty-two gallons per head per day is reduced eventually to twenty-one gallons, you will require an engine to pump 2,400 gallons per minute for ten hours per day for six days per week, thus avoiding working on a Sunday or at night. Then, with regard to the construction of engine to do this work in a properly economical manner, I am decidedly of opinion that the Worthington type of engine is the right one for your purpose, as it is highly economical in working, due to a large amount of expansion. It is easy to manage, and moderate in first cost, besides which it does not require a massive and expensive engine house, as most beam engines do, and it is not likely to get out of order, being very simple in its construction. There are now many engines of this construction in this country, several of which I have examined, and one of which I worked for a twenty-four hours' trial, and there are a very large number in America, where they were first introduced many years ago. One very great advantage is that the engine can be worked either slow or fast. Thus you need not increase your filter beds until the quantity of water required necessitates your doing so." The Corporation accordingly instructed Mr. E. K. Burstal to prepare a specification for a Worthington engine to be constructed by Messrs. James Simpson & Co., of London, who are the sole licensees for Messrs. H. R. Worthington's patents, and this engine has been at work for about three months, and has given the most unqualified satisfaction.

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* From *The Engineer*, March 6th, 1891.

Mr. W. H. White, C. E., the engineer of the water works, made a careful trial of this engine, assisted by Mr. Robert Downing, C. E., the chairman of the Water Works Committee, and obtained the following results:

Date of trial,	January 15, 1891.
Duration of ditto,	6 hours,
Total number of revolutions,	7301
Average revolutions per minute,	20.25
Average total lift,	185.86 ft.
Total pounds of feed water, including jackets,	12,408
Pounds of feed per hour,	2,068
Total pounds of coal, gross, including ash,	1,213
Gallons pumped per revolution,	100
Pump horse-power $= \frac{100 \times 10 \times 185.86 \times 20.25}{33,000} =$	113.9
Indicated horse-power—mean of 48 diagrams,	123.34
Pounds of feed per pump per horse-power per hour $= \frac{2068}{141}$, including jackets,	18.14
Pounds of coal per pump horse-power per hour $= \frac{1213}{6 \times 114} =$	1.77
Evaporation, pounds of water per 1 pound of coal,	10.2
Efficiency, $\frac{\text{Pump horse-power}}{\text{Indicated horse-power}}$	92.3 per cent.
Duty, assuming evaporation of 10 pounds of water per pound of coal, as required by contract, =	122,000,000
Duty on actual coal burnt. =	125,100,000

It will be observed that the pounds of coal as actually used represent a duty of 125 millions, while 122 millions is the duty on an assumed evaporation of 10 to 1, so that the duty actually obtained is almost exactly the same as that calculated. These results must be acknowledged as exceedingly good. The engine works in the most perfect manner, and the Corporation of Oxford are to be congratulated on having such a valuable increase to their pumping power.

HAMMERSMITH, ENGLAND.

7,000,000 GALLONS CAPACITY.

ONE ENGINE.

REPORT OF ROBERT CHADWICK, C. E.*

THE other illustration is of the 220 horse-power Worthington engine erected at the West Middlesex Water Works, Hammersmith. This engine, it will be seen, is rather differently designed from the Oxford one, as it had to be specially arranged to go exactly on to the existing foundations of two Cornish engines which it has replaced. This necessitated a different arrangement of driving the valve and air pumps to that which is usually adopted; and as there are more moving parts, the efficiency or friction of the engine is slightly in excess of that usually obtained by the Worthington engine on similar lifts. This will be seen on reference to the table at the end of this article, the loss being about 4 per cent. The efficiency, however, under these circumstances, is 87 per cent., or in excess of any type of fly-wheel engine when the valves and pump plungers are tight; in fact, the Worthington engine is in this respect far superior to a fly-wheel engine, as if there is any slip on the pump valves or in the plungers, the engine shows it immediately and does not work well.

Careful experiments show that the slip on Worthington engines, even after they have been at work for long periods of years, to be almost inappreciable, say 1 to 1¼ per cent.; in fact, the Worthington pump is practically a water meter. Two trials of this engine were made by Messrs. James Simpson & Co., who found the feed-water, exclusive of the jackets, to be 16.1 pound and 16.3 pound per pump horse-power per hour, which is almost similar to the result obtained by Mr. E. A. Cowper on a 24-hours' trial of the engine at the New River Water-Works. For the purpose of ascertaining if the contract conditions were fulfilled, the West Middlesex

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* From *The Engineer*, March 6th, 1891.

Water Works had a 24-hours' trial made by Mr. Robert Chadwick, C. M. G., C. E. This was very carefully carried out, and the following record is most complete, and exceedingly interesting on account of the varying head that occurs in the service mains, as can be seen from the diagrams of pressures. The differential accumulator invented by Mr. C. C. Worthington, whereby the pressure in the compensators adjusts itself direct from the pressure in the mains acts very satisfactorily, rendering the engine perfectly automatic in every way, and Mr. Chadwick's remarks on this differential accumulator and the experience of Mr. M. W. Hervey, the engineer to the West Middlesex Water Works Company, when a burst took place in the main, are interesting, and are worthy of careful attention.

The following is an account of a 24-hours' steam trial of a compound Worthington high-duty pumping engine constructed by Messrs. James Simpson & Co. for the West Middlesex Water Works, on November 17th and 18th, 1890:

GENERAL DESCRIPTION OF THE PUMPING ENGINE.

The engine is of the ordinary Worthington type, and consists of a pair of horizontal compound engines placed side by side, each working a double-acting pump. There is no crank shaft or flywheel. Each engine works the main valves of its fellow, but the expansion valves, which are attached to both high and low pressure cylinders, are worked direct by the engine to which they belong. The variations in the steam pressure on the pistons, due to the high grade of expansion employed, are equalized by means of compensating cylinders, which form the "high-duty attachment," an essential feature in this engine, the action of which will be described in detail later on. The main steam valves are of the Corliss type. Two are provided to each high-pressure cylinder, and serve both for steam and exhaust. The low-pressure cylinders are each provided with four valves, two for steam above and two for exhaust below. The expansion valves consist of two plates, working on a fixed seat in the valve chest. The grade of expansion is variable by hand. The high-pressure cylinders are placed next to the pumps and have a single piston rod, whilst the low-pressure cylinders have each two rods, passing outside the high-pressure cylinders. The three rods are connected together, and to the pump-rod, by a cross-head. The pumps are connected to the cylinders by means of a strong cast-iron frame, so that the whole forms one rigid and self contained mass. The pumps are bolted down to the foundations, but the cylinders are not, and consequently quite free to move by expansion or contraction. To these connecting frames are attached the oscillating or compensating cylinders which form the "high-duty attachment," which permit, as already stated, the use of high grades of expansion. The pumps are of the usual Worthington type, consisting of two chambers having a common partition wall, in which there is a collar or sleeve, through which a solid plunger works, the pump valves being multiple as is usual. The engines are provided with a jet condenser, and each engine has a pair of air pumps, which, as space is limited, are arranged beneath it. These air pumps are worked by a lever and rock shaft, which gives motion to the steam valves. The foundations are those on which two Cornish beam engines had been at work for many years, and were scarcely altered when the Worthington engine was put down. The general action of the engine resembles that for a pair of engines connected to a common crank shaft, with cranks at right angles. There are, however, no cranks or connecting rods, so that the length of the stroke is not controlled by any positive mechanism, but only by the adjustment of the valves and of the oscillating cylinders, or high-duty attachment already mentioned. As the pistons are not constrained to move with definite relative velocities but work independently, each taking up the work of the other as it approaches the end of its stroke, the flow of water is more regular than in the case of a pair of coupled fly-wheel engines.

LEADING DIMENSIONS.

Diameter of high-pressure cylinders,	22	in.
" " low-pressure cylinders,	43	"
" " high-pressure piston-rod, one on one side,	4	"
" " low-pressure piston rod, two on one side,	4	"
" " pumps A,	19.485	"
" " pumps B,	19.492	"
" " pump-rod, one on one side	4¼	"
Common length of stroke for cylinders and pumps,	48	"
Extreme length of stroke possible from cover to cover of cylinder,	49	"
Diameter of oscillating compensating cylinders, two to each engine,	8	"

Hence:—

Mean effective area of high-pressure cylinder,	372.82	sq. in.
" " " low-pressure cylinder,	1,439.63	"
" " " pump plunger,	291.205	"

Area of oscillating cylinders, two to each pump, -	50.27 sq. in.
Pounds per square inch on pump plunger—per pound per square inch effective on high-pressure cylinder,	1.284 lb.
Pounds per square inch on pump plunger—per pound per square inch effective on low-pressure cylinder,	4.943 "
Ratio of area of high and low-pressure pistons, - -	3.851
Gallons displaced per foot of piston travel, - - -	12.603

The diameter of the pump plungers was measured by Professor Unwin from gauges prepared by Mr. Hervey, C. E., the engineer of the West Middlesex Water Works. The other dimensions are taken from the manufacturer's drawings.

MANNER OF EXPERIMENTING.

The trial lasted from 10 A. M. on Monday, November 17th, to 10 A. M. on Tuesday November 18th. Between 8.30 A. M. and 9.30 A. M. on Monday, the fires were cleaned and clinkered, so that by 10 o'clock they were bright and clear. The depth of coal on the grates was then estimated, the height of water in the gauge glasses noted, and the trial commenced. At or about the same time on Tuesday morning the fires were again cleaned and made up, so that at the conclusion of the trial—10 A. M.—the fires were clean and bright, and practically in the same condition as at the commencement; and the coal consumed in restoring fires is therefore included in the total consumption. The water in the boilers also was brought back at the end of the trial to the same level in the gauges as at its commencement. The engine under experiment was performing the normal work of the water works. It pumped into the general system of mains in common with several other Cornish and rotative engines, which were stopped and started from time to time during the day according to the demand. The water pressure oscillated continually, and was subject to great and sudden variations, due to the varying draught on the main. A statement of the working of the other engines is appended. The coal was weighed in sacks, each nominally containing 2 cwt. The weighments were checked by hanging stamp weights of 2 cwt. to the arm of the steelyard, to which the sacks were suspended in weighing. The coal used was "Nixon's Navigation." It was wet, and considering the large percentage of ash, cannot be said to have been of very excellent quality. The feed water supplied to the boiler was measured by means of a tank. It was taken from the mains, and not from the hot-well. The tank had a sharp-lipped overflow pipe above, and the discharge pipe below was also turned up and terminated in a circular orifice with a thin edge; so that exactly equal tankfuls of water could be discharged, as required, into a cistern below, whence the donkey engine pumped the water to the boilers. The exhaust steam from the donkey engine was passed into the cistern, whence it drew the feed water, being completely condensed and heating it to a certain extent. The whole of the heat rejected by the donkey engine in its exhaust steam was therefore returned to the boiler. The precise amount of heat, corresponding to the work done in pumping the feed into the boiler, is converted into work. None is rejected. In the present instance the donkey pumped the feed from the mains at a temperature of 50 degrees, instead of from the hot-well at about 70 degrees, so that every pound of water had to be raised 20 degrees more than it would have been if taken from the hot-well. This occasions a loss, as will be shown later on. The jacket drains returned to the boilers, so that the feed water measured represents the amount of steam passing through the cylinders only, exclusive of that condensed in the jacket.

The principal object of the trial was to determine the coal consumption, so that it was thought desirable to disturb the normal conditions of working as little as possible. The contents of the measuring tank were ascertained by weighing twice, with reversed arms of the balance, and the two weighments, which were made by about 90 pounds at a time, agreed to 1 ounce. The time of delivering each tankful of feed water, as well as the total number used, was recorded by observers. To check the same, a counter was provided, worked by a float, which rose and fell with the water in the measuring tank. The measurement of the stroke of the engine, which, as I have said, is not limited by any positive mechanism, formed an important factor in the experiment. To obtain this with sufficient precision, scales, graduated into inches and tenths, were attached to the frames of the engine. A pointer, fixed to each cross-head, marked the position of the pistons with regard to their central position on these scales, at or near to the end of their stroke. Two observers noted the position of the pistons as they paused for an instant at the ends of 100 consecutive strokes. This was done every fifteen minutes alternately, first on one engine and then on the other. The mean of these 9,600 measurements was taken as the mean stroke. The measurements were made at regular intervals, without selection of time, or special adjustment of the stroke. The pressure in the mains, against which the pumps worked, was measured by means of a mercurial pressure gauge. The suction lift was determined by means of a float and scale. Both were recorded at intervals of 15 minutes. The mean reading of the pressure gauge, which was graduated in feet of water, was checked by actual measurement in inches, and found to agree very nearly with the scale reading. The actual height of the mercurial column, corresponding to the mean reading of the gauge is, however, used in the principal calculations. In the following calculations, although there are no revolving parts, the expression "revolution" is used in the sense of a pair of complete strokes of each engine, the cycle corresponding to a revolution of the crank shaft of a similar pair of rotative engines. The expression "pump horse-power" signifies the space swept through by pistons per minute, multiplied by the pressure on the pump piston in pounds, and divided by 33,000. It is therefore the work done on the water by the engine, exclusive of friction of engine and pumps.

RESULTS OF THE TRIAL.

Total number of revolutions made in 24 hours, - -	28,917
Therefore mean number of revolutions per minute, -	20.081
Mean mercurial column in water-pressure gauge, - -	154.8 in.
Mean depth of surface water in suction well below the zero of mercurial gauge, in feet, - - - - -	8.74 ft.
Temperature of mercury in pressure gauge, - - -	72° Fahr.
Temperature of water in mains, - - - - - -	50°

Therefore, pressure on pump plunger, in pounds per square inch—

$$\left\{\frac{154.8}{12\times 144}\times\frac{848.75\text{ lb.}}{10.0001}(70^\circ-32^\circ)\right\}+\left\{\frac{8.74\text{ ft.}}{144}\times 62.409\text{ lb.}\right\}$$
$$=79.52\text{ lb.}$$

Mean length of stroke, in inches, - - - - -	48.14 in.
Space swept through per revolution, in feet, -	16.05 ft. nearly

Hence, mean pump horse-power developed =

$$\frac{\overset{\text{Rev.}}{28{,}917}\times\overset{\text{ft.}}{16.05}\times\overset{\text{area.}}{291.205}\times\overset{\text{lb.}}{79.52}}{24\times 60\times 33{,}000}=226$$

Gross weight of coal consumed, - - - - -		10,987 lb.
Ashes weighed, - - - - - - -	232 lb.	
Clinker weighed, - - - - - -	412 lb.	
Together, - - - - - - - - - -		644 lb.
Hence, net fuel burned, - - - - - - -		10,343 lb.
Gross pounds of coal per pump horse-power per hour,		2.03 nearly

Net fuel burned per pump horse-power per hour, - 1.91 nearly
Duty in millions, - - - - - - - - 116,100,000
Number of measuring tanks of water supplied to boiler in 24 hours, - - - - - - - $161 + \frac{9.5}{24}$
Contents of measuring tank in pounds, - - 544 lb. + 10.5 oz
Hence, total feed-water supplied to boiler, - - 87,906 lb.
Therefore, weight of steam through cylinders per pump horse-power per hour $\doteq \frac{87.906}{24 \times 226} =$ 16.2 lb. nearly
Steam through cylinders per stroke = .76 lb.

Numerous diagrams were taken during the trial, which gave the following results:—

h = mean effective forward pressure in high-pressure cylinders, - - - - - - - - - 34.07
= mean effective forward pressure in low-pressure cylinders, - - - - - - - - - 9.55
H = Effective pressure per square inch of pump due to steam pressure in high-pressure cylinder, - - 43.74
L = Effective pressure per square inch of pump due to steam pressure in low-pressure cylinder, - - - 47.22
S = H + L = Total working pressure of steam per square inch of pump piston, - - - - - 90.96
Water pressure in pounds per square inch, - - 78.92

$E = \frac{W}{S} =$ Efficiency of engines, - - - - - 0.87

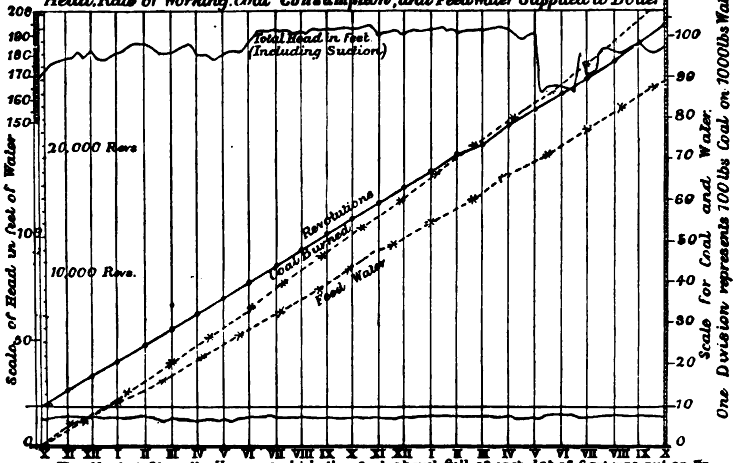


Diagram Shewing Head, Rate of Working, Coal Consumption, and Feed Water Supplied to Boiler

The Marks * Show the Hours at which the first shovel full of each lot of 6 cwt was put on Fir
The Marks * Show the Hours at which Each Tenth Measuring Tank was emptied.

A subsequent experiment to test the efficiency of the engines, made on the 1st of January, gave practically the same result, so that the loss of energy by internal friction of the engine and pumps is 13 per cent. nearly. The average steam pressure was about 80 pounds above the atmosphere, corresponding to a temperature of about 324° Fahr. Approximate temperature of hot-well, 70 degrees; the total heat of formation of 1 pound steam at 324 degrees from 32 degrees = 1180.8; less 32 degrees to 70 degrees = 38; the total heat of formation from 70 degrees 1142,8 The total heat therefore supplied to the cylinders of the engines per pump horse-power developed is $\frac{16.2 \times 1142}{60} = 308.5$ T. U. per pump horse-power. In addition to the heat supplied to the steam which passes through the cylinders, there is that which is given up by the steam that condenses in the jackets. This may be taken at about 15 per cent. of that passing through cylinders, and giving up its latent heat only. Hence heat due to condensation in jackets $\frac{16.2 \text{ lb.} \times 886 \text{ T. U.}}{6.67 \times 60} = 36$ T. U. Therefore T. U. supplied to engine per pump horse-power per minute $= 308 + 36 = 344$ T. U. But it appears that the indicated horse-power exceeds the pump horse-power in the ratio 0.87 : 1.00. Then T. U. per indicated horse-power $= 344 \times 0.87 = 299.28$ T. U. Hence, efficiency of steam engine $= \frac{42.75}{299.28} = .143$.

The correction for the cold feed-water may now be determined. Each pound of feed-water had to be raised from 50 degrees to the boiling point, corresponding to the steam pressure, instead of from 70 degrees, the temperature of the hot-well. Total heat of formation of 1 pound steam at 95 pounds absolute from 50 degrees = 1162 T. U. To raise 1 pound from 50 degrees to 70 degrees = 20 T. U., or 1.72 per cent., in addition to that required to raise steam from 70 degrees, an amount that should be deducted from the coal consumption had the donkey drawn from the hot-well.

Per contra, however, the engine in working the feed-pump would have to develop a greater horse-power, and would, consequently, require more heat than under the actual conditions. This account may be easily calculated, as it happens that the boiler pressure, 80 pounds above the atmosphere, is very nearly equal to the water pressure against which the pumps work. The feed amounts to 61 pounds, or 6.1 gallons per minute, and the total water pumped to about 4,000 gallons a minute. So that if the feed-pump were worked by the engine, it would do an amount of work equal to 4,006 gallons a minute, instead of 4,000, or 0.15 per cent. in addition. Deducting this from the loss 1.72, we have a total loss of 1.57 per cent. The reduced coal consumption therefore becomes $2.03 \times \frac{98.43}{100} = 1.998 = 2.00$ nearly pounds per horse-power per hour, or $1.91 \times \frac{98.43}{100} = 1.88$ nearly pounds per horse-power per hour.

Duty = 118,000,000 millions.

The efficiency of a perfect heat engine, between the same limits of temperature, is

$$\frac{(461 \text{ deg.} + 324 \text{ deg.}) - (461 \text{ deg.} + 70 \text{ deg.})}{(461 \text{ deg.} + 324 \text{ deg.})} = 0.323$$

The relative efficiency to a perfect heat engine is, therefore $\frac{0.143}{0.323} = 0.442$ nearly.

DESCRIPTION OF HIGH DUTY ATTACHMENT.

The water pressure varied considerably at every instant during the trial. Not only was there a marked oscillation, due to the intermittent action of the Cornish engines, working into the same system of mains, but there were also sudden changes of pressure, much greater in extent, due to the opening and shutting of valves, conditional on the intermittent supply which is given in a large part of the district. Daily, at about 4.45 A. M., a sudden fall of about 30 feet takes place in about forty seconds, due to the supply being turned on to a lower level service reservoir. Nevertheless, the high duty attachment adjusts itself to these variations, so that no inconvenient variation of stroke is produced by them. Indeed, Mr. Hervey related that on one occasion, whilst this engine was at work, a 30-inch main burst within three and a half miles of the engine house. The pressure fell rapidly, about 80 feet, but no inconvenience of any kind was experienced. The action of the high duty attachment, and the manner in which it is adjusted automatically to meet variations in pressure, is as follows: In a single cylinder steam engine working expansively, the forward pressure exerted by the steam at any portion of the stroke may be represented with sufficient accuracy by the ordinates of the diagram, Fig. 1, P T S, R Q. The propelling force of the steam, therefore, represented by the ordinates P T, P_1 S_1, P_2 S_2, etc., varies at each movement of the stroke at P Q. The resistance of the pump with its friction is nearly constant, and may be represented by the horizontal line M N coinciding with the mean forward pressure of the steam. At the commencement of the stroke, therefore, the steam pressure is in excess of the resistance in the latter part, in defect of the same. Consequently when the piston reaches the point C, corresponding to the mean forward pressure, an amount of energy, represented by the area M T S C, must be stored up, and restored again during the second part of the stroke to make good the deficiency, which is represented by the area C N R, otherwise when the piston reached the point in its path corresponding to C, where the steam pressure is equal to resistance, there would be equilibrium, and motion would cease. In engines with fly-wheels the surplus energy in the first part of the stroke is stored up, in the form of kinetic energy, in an acceleration of the fly-wheel. In the case of Cornish engines the heavy beam and counter-weights perform, to some extent, the same duty as the fly-wheel. In direct-acting engines, where the moving parts are relatively light, the pump being connected direct to the steam pistons, the moving mass is not sufficient to store up and restore the necessary energy, unless, indeed, the pistons are allowed to attain a speed improper for pumps. Consequently in a single cylinder direct-acting engine, no appreciable grade of expansion can be employed unless a fly-wheel is introduced. The sum of the mean forward pressures of the two cylinders of a compound engine, when combined and reduced to

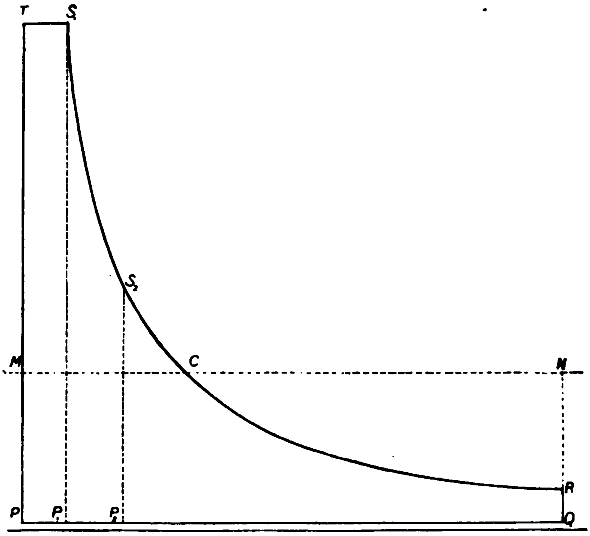


FIG. 1.

their resultant pressure on the pump-rod, is more regular than that of the single cylinder. When actual indicator diagrams are thus treated, the effect of clearance, wire-drawing, irregular back pressure, and intermediate spaces are such that the line of combined steam pressure is found to be an S-like curve, like S C R, Fig. 2. When the grade of expansion is moderate, say 2 to 3 only, this curve coincides sufficiently nearly with the horizontal line of mean pressure to admit of a full stroke, the inertia of the moving parts making up the differences. To enable as high a grade of expansion to be used in a non-rotary direct-acting engine, and with the same advantage as in a fly-wheel engine, some arrangement must be adopted for storing up and restoring the surplus energy, thus equalizing the steam pressure throughout the stroke. In the present instance this is done by a pair of small oscillating cylinders, Fig. 3, connected to the piston-rod of each engine, which form what has been called the "high-duty attachment." The trunnions of these small oscillating cylinders, or "pots," as they will be called for brevity, are carried in bearings in the main framing. The pots

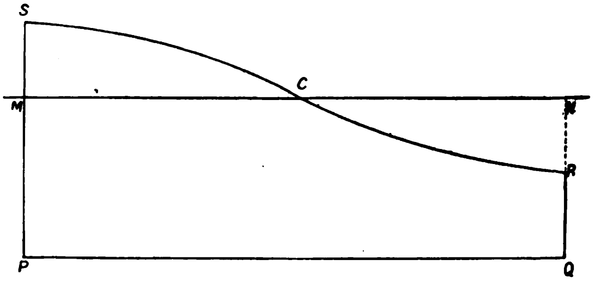


FIG. 2.

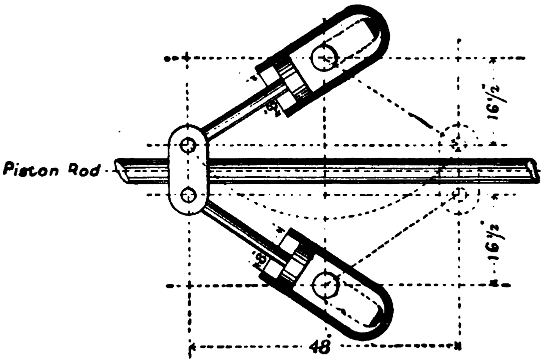


FIG. 3.

are closed at the bottom and provided with pistons, the rods of which are jointed to the main piston-rod. One trunnion is hollow, and connected to the interior of the pot, near the closed end, by a passage cast in the side thereof. The hollow trunnion is in connection with a receiver containing water at high pressure, which fills the pots beneath the piston. The water in this receiver is maintained, as far as each individual stroke is concerned, at a constant pressure of 200 pounds per square inch. The pot pistons, therefore, are exposed to a constant pressure of about 200 pounds per square inch tending to force them out of the pots. The pots oscillate at each stroke through an angle of rather more than 90 degrees. During the first half of the stroke the pot pistons are pushed in and displace the contained water, forcing it into the receiver, doing work, say, by raising a weight like that of an ordinary accumulator, thus storing up energy. At the middle of stroke the inward movement of the pot piston ceases and an outward movement commences; the water then re-enters the cylinders from the receiver, and gives out energy in the direction of the movement of the steam piston, thus helping it forward, and restoring the energy stored up in lifting the weight during the first half of the stroke. The pots being equal and in pairs, the components of their piston pressure, at right angle to line of motion of the piston, are in equilibrium and need not be discussed. It will also suffice to consider

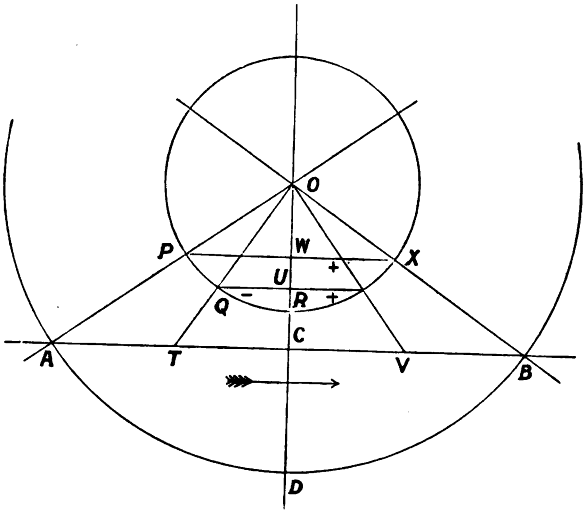


FIG. 4.

the pressure of one pot, doubling its amount for the pair. Let A B, Fig. 4, be the path swept through by the pin connecting the pot piston-rod to the main piston rod. Let O be the centre of oscillation of the pot, situated in a line perpendicular to A B, through its central point. Draw the arc A D B. Then it is evident that the stroke of the pot pistons is equal to the versine C D. From this we can at once calculate the total work done in forcing in the pot piston during the first half of the stroke, and given out again in the second half of the same.

Let a = area of a pair of pot pistons.
P = the pressure on the same per square inch.
Then $P\,a$ = total pressure on a pair of pot pistons.
Let S = stroke of pots.
The work done $= P\,a\,s$.

$$\text{But } S = \overline{A\,O} - \overline{O\,C} = \overline{D\,O} - \overline{O\,C}$$

$$= \overline{D\,O} - (\overline{A\,O^2} - (\overline{A\,C^2})\tfrac{1}{2}.$$

$$\text{But } \overline{A\,O^2} = \overline{A\,C^2} + \overline{C\,O^2}$$

$$\text{And } D\,O = A\,O = \sqrt{\overline{A\,C^2} + \overline{C\,O^2}}$$

$$\text{Hence } S = \sqrt{\overline{A\,C^2} + \overline{C\,O^2}} - \overline{C\,O}$$

Let $A\,C = h$ and $C\,O = d$

$$S = \sqrt{h^2 + a^2} - d.$$

In this case $h = 2'$, $d = 1.375'$ hence $S = 1.094'$
Also $P = 200$ lb., $A = 100$ nearly, hence $W = 1.094 \times 20{,}000$
$= 21{,}880$ foot-lbs.

If we divide this by the half-stroke 2′ we obtain the mean or average pressure, opposing the main piston during the first half of its forward stroke, and assisting it during the second half, or

$$\frac{21{,}880}{2} = 10{,}940 \text{ lb.}$$

As the area of the high-pressure piston is 374 square inches nearly, this is equivalent to an average pressure of about 29 pounds per square inch deducted from the forward pressure during the first half of the stroke and added to the pressure during the second half.

So far the mean effect of the oscillating cylinders has been considered. Next, to examine the pressure which they exert at each successive point in the stroke. To do this, it will be convenient to reduce their action to its equivalent in pounds per square inch on the high-pressure cylinder. Let A = area of the high pressure cylinder, and let p be the equivalent pressure exerted thereon by the pot pistons, when horizontal or parallel to the main piston.

$$\text{Then } p = \frac{a P}{A},$$

$$P = 200 \text{ lbs., then } p = \frac{200 \text{ lbs} \times 100}{375} = 53 \text{ lbs. nearly, or if}$$

$$\text{P} = 166 \text{ lbs., then } p = \frac{166 \times 100}{375} = 44 \text{ lbs. nearly.}$$

Referring to the diagram, Fig. 4, with O as a centre describe the circle $P\ Q\ R\ S\ X$, and radius representing to any convenient

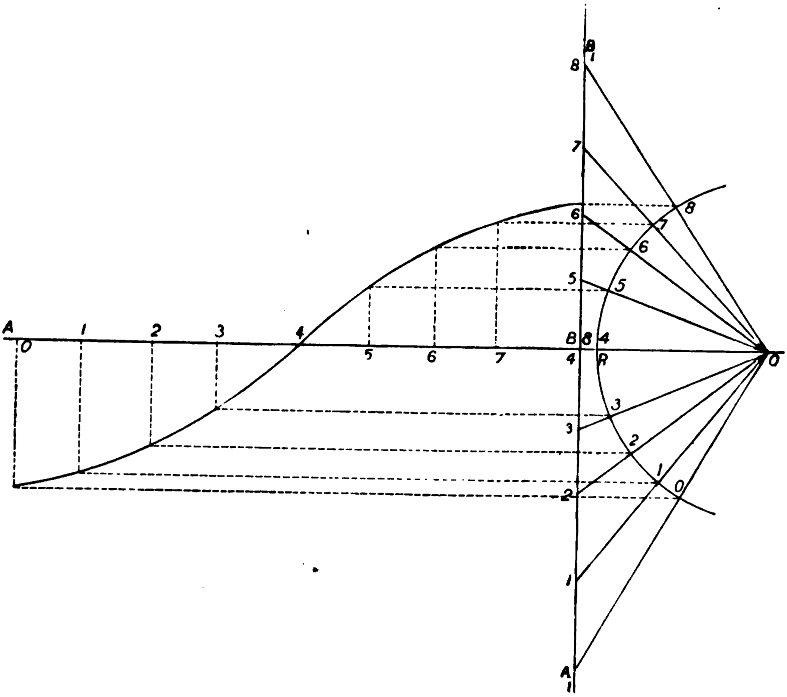


FIG. 5.

scale the pressure p, which is constant. Then it is clear that the component of the pressure p, parallel to the line of the main piston-rod $A\ B$ is, at the commencement of the stroke, equal to the perpendicular $P\ W$ measured on the scale used for p, in drawing the circle, whose radius represents the constant pressure. At any other points of the stroke such as T and V, the perpendicular, or lines $Q\ V$, $V\ S$, represent the resultant pressures parallel to the main piston-rod, respectively. The cosines $O\ W$, $O\ U$, represent the pressures perpendicular to the rod, which are balanced by those of the opposite pot of the pair.

At the centre of the stroke C the resultant is zero. The resultants to the left of the centre line are negative, and represent pressures opposing the piston; whilst those tc the right are positive, and represent pressures assisting the motion of the piston, which, in both cases, is supposed to be in the direction of the arrow, from left to right. On the back stroke the signs will be reversed. The actual pressure at each instant can be conveniently determined graphically as follows: Make $A\ B$, $A_1\ B_1$, Fig. 5, each equal to the length of stroke to any convenient scale, and make $A\ B$ bisect $A_1\ B_1$ at right angles and prolong to o. Make $B\ O$ to same scale equal to h, the distance of pot-axis from line of motion of the connecting point of the pot piston-rod with the main piston-rod. Graduate both $A\ B$ and $A_1\ B_1$ similarly. With radius $O\ R$ and centre O, equal to the pot-pressure p, draw a circle. Draw rays from the centre O to the various graduation of $A_1\ B_1$. From the points where these rays cut the circle draw parallels to $A\ B$, cutting ordinates drawn through the corresponding graduations in $A\ B$. Joining the points thus found a curve is formed whose ordinates represent the pressure exerted by the pots in the line of motion of the main piston-rods. The ordinates below the base-line are negative, and represent pressures opposing the motion of the main piston from left to right. Those above the same are positive, and represent pressure assisting the motion of the main piston. The curve thus obtained resembles the projection of a spiral, or a harmonic curve. It differs from it in that the two branches have two asymptotes, parallel to the base-line and touching the generating circle for the generating circle. It will be observed that whatever radius be chosen for the generating circle, or, in other words, whatever be the pot-pressure, the form of a curve will always be similar, the area above and below the base-line, or the work done against or with the piston's motion being directly proportional to the pressure in the pots only. It will also be apparent that the curve of pot-pressures resembles that of the combined forward pressure of the expanding steam; and by selecting a proper pressure in the pots it may be made to coincide closely with it when reversed. By combining the positive and negative pressures of the pot diagram with the steam diagram a final line of resultant forward pressure may be obtained, differing so little from a horizontal line that the difference may be adjusted by the inertia of the moving parts.

The diagrams given in Figs. 6 and 7 show the actual application of these principles to indicator diagrams taken at 2.15 A. M. and

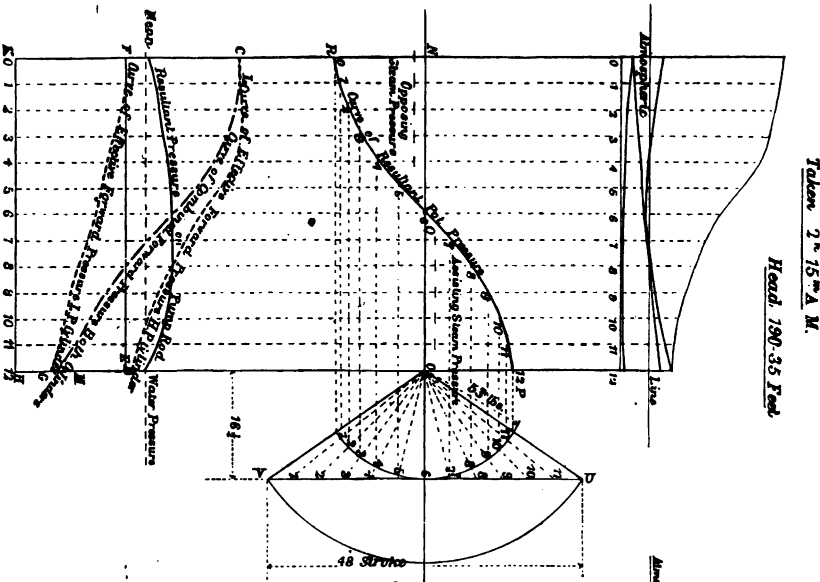


Fig VI
Mean Diagram
Taken 2h 15m A.M.
Head 190·35 Feet

Atmospheric Line
Curve of Resultant Pot. Pressure
Opposing Steam Pressure
Assisting Steam Pressure
55 lbs.
Curve of Effective Forward Pressure H.P. Cylinder
Curve of Combined Forward Pressure on Pump Rod
Resultant Pressure
Mean
Water Pressure
Curve of Effective Forward Pressure L.P. Cylinder
Curve of Combined Forward Pressure Both Cylinders
16¼
48 Stroke
V
U
M
O
P
R
C
F
G
H
K

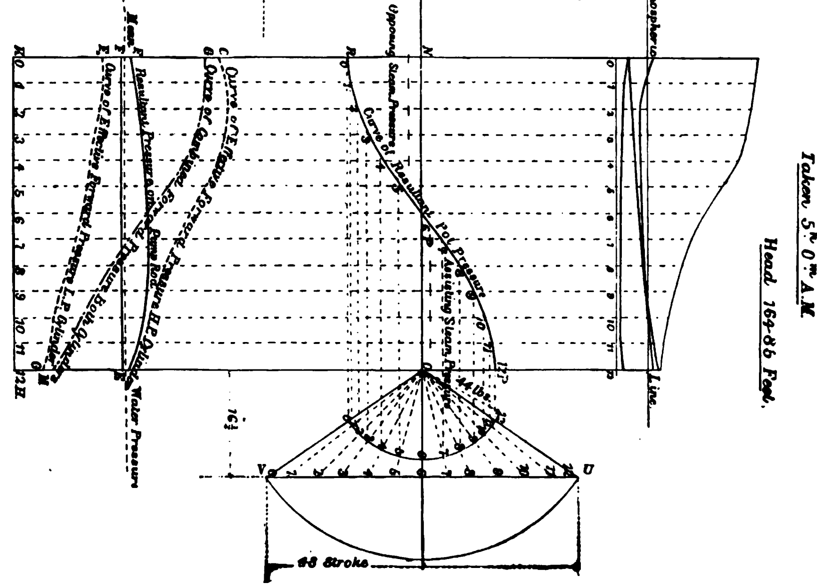


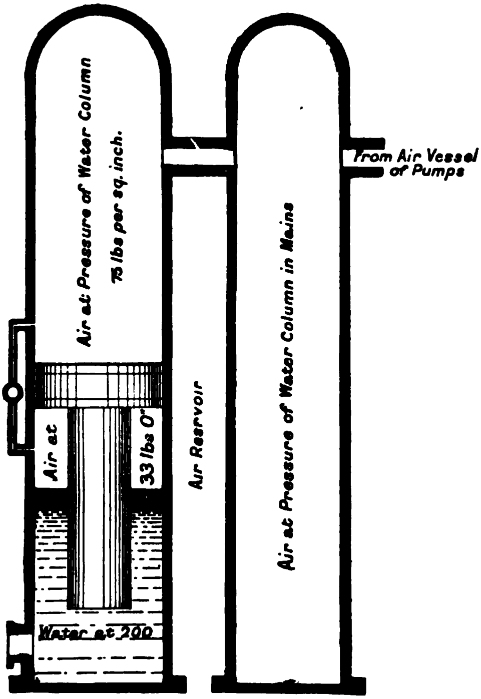
Fig. VII
Mean Diagram
Taken 5h 0m A.M.
Head 164·86 Feet.

Atmospheric Line
Curve of Resultant Pot. Pressure
Opposing Steam Pressure
Assisting Steam Pressure
44 lbs.
Curve of Effective Forward Pressure H.P. Cylinder
Curve of Combined Forward Pressure on Pump Rod
Resultant Pressure
Mean
Water Pressure
Curve of Effective Forward Pressure L.P. Cylinder
Curve of Combined Forward Pressure Both Cylinders
16¼
48 Stroke
V
U
M
O
P
R
C
F
G
H
K

5 A. M., when the water pressure in the mains was high and low, respectively. In both figures, *A* and *B*, are the mean diagrams

from those of the two sides of the pistons, both engines drawn to the same scale. *C D E F* is the diagram of effective forward pressure in the high-pressure cylinders, back pressure being deducted from the forward pressure, taking care to invert the back pressure line. *K F G H* or *K F*, *G H*, is the diagram of effective forward pressure in the low-pressure cylinder, obtained in the same manner, but its ordinates enlarged in the ratio of the area of the cylinders to represent the equivalent pressure on the high-pressure cylinder. *K C L M H* or *K C*, *M H*, are the diagrams of total effective forward steam pressure, made by adding together the respective ordinates of the high and low-pressure diagrams. *R Q P* is the diagram of the pots, drawn as described with the pot-pressure observed at the time. Then, by setting the ordinates of this curve down from the curve *C H* in the first half of the stroke, and upwards in the second, according to the signs, the final curve of resultant forward pressure is obtained, which, though irregular, differs much less from a straight line than either of the original curves.

When an engine has to pump against a steady load, then all that is required is to adjust once and for all the pressure on the pots to the amount requisite to produce a full stroke, by placing a sufficient weight on the ram of an accumulator connected with the oscillating cylinder. But with an irregular varying load this arrangement would not suffice. If a reduction of pressure took place so suddenly that the steam valves could not be re-adjusted to meet it, the energy stored up in the pots during the commencement of the stroke would be in excess of the amount required to complete it, and the piston would probably strike the ends of the cylinder. To meet this difficulty the pressure on the pots is made to vary with the water column on the pumps. The pressure of the water in the mains is too low for direct application, as it would necessitate pots of inconveniently large diameter. It is therefore multiplied and adjusted in the following manner by means of a differential accumulator, invented and patented by Mr. C. C. Worthington. A large receiver is connected with the air space of the air vessel connected with the main pumps, and contains compressed air at the pressure of the water column. This is admitted to the upper part of a cylinder provided with a piston. A ram, having a diameter equal to one-half that of the piston, and consequently one-fourth of the area, is attached to the same, and passes through a stuffing-box at the bottom of the cylinder into a second cylinder which contains water, and which is in connection with the pots. The pressure of the air corresponding to the water column produces a four-fold greater pressure on the pots. As this would be greater than necessary, and in order to permit of adjustment of the pot-pressure to meet variations in steam pressure, and in the grade of expansion, a certain amount of compressed air is admitted to the annular space below the piston and maintained at a lower pressure than that due to the water column. The pressure in the pots is, therefore = (the water column × 4) – (the intermediate pressure × 3); or, in practice about (75 lbs. × 4) – (33.3 lbs. × 3) = 200 lbs. By admitting air from above the pistons to the annular space below it, the pot-pressure may be reduced till it equals that of the water column. By allowing the air to escape from the annular space, the pot-pressure may be increased till it equals four times that of the water column. Between these limits any desired pressure can be established in the pots. The effect of this is that the pot-pressure is maintained at some constant ratio to that of the water column When the water column falls the pot-pressure is reduced, and *vice versa*. By interposing a large vol-



DIFFERENTIAL ACCUMULATOR.

ume of air between the accumulator piston and the water column, a certain elasticity is given to the whole arrangement, and the air, owing to its small density, offers less resistance to a sudden change than a body of water. The effect of a sudden reduction of pressure is immediately to shorten the length of stroke, the number of strokes being increased. An air pump is provided to replenish the air vessel. The loss of air is, however, insignificant. During the trial it was only worked for about twelve minutes, and during the twenty-two hours' working on the Friday and Saturday preceding the trial the air pump was not worked.

GENERAL ACTION OF STEAM IN CYLINDERS.

As the amount of steam condensing in the jackets was not measured, the data for an examination of the action of the steam at different points of the stroke are not available. In Fig. 9, the mean diagrams, compiled from seven cards from the high and low-pressure cylinders, are combined; the low-pressure diagram being shown as though the expansion took place in two cylinders having the same diameter as the high-pressure cylinder, one having the same length of stroke as the high-pressure cylinder, and the other a longer stroke, so that their joint capacity is equal to that of the actual high-pressure cylinder. A saturation curve is drawn through

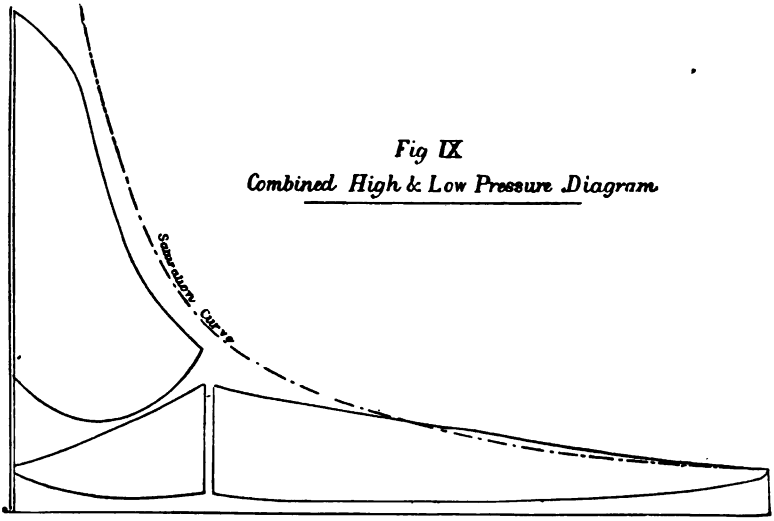


Fig IX

Combined High & Low Pressure Diagram

a point representing the volume of the mean weight of steam passing through the cylinder per stroke, 0.76 pound. This shows pretty clearly that considerable re-evaporation takes place as the stroke advances. But it is to be remembered that the indicator diagram does not necessarily represent the volume of the same weight of steam at different points of the stroke. The clearance and intermediate spaces are not great, but no information exists as to the quantity of steam or water which they contain at the different phases of the stroke.

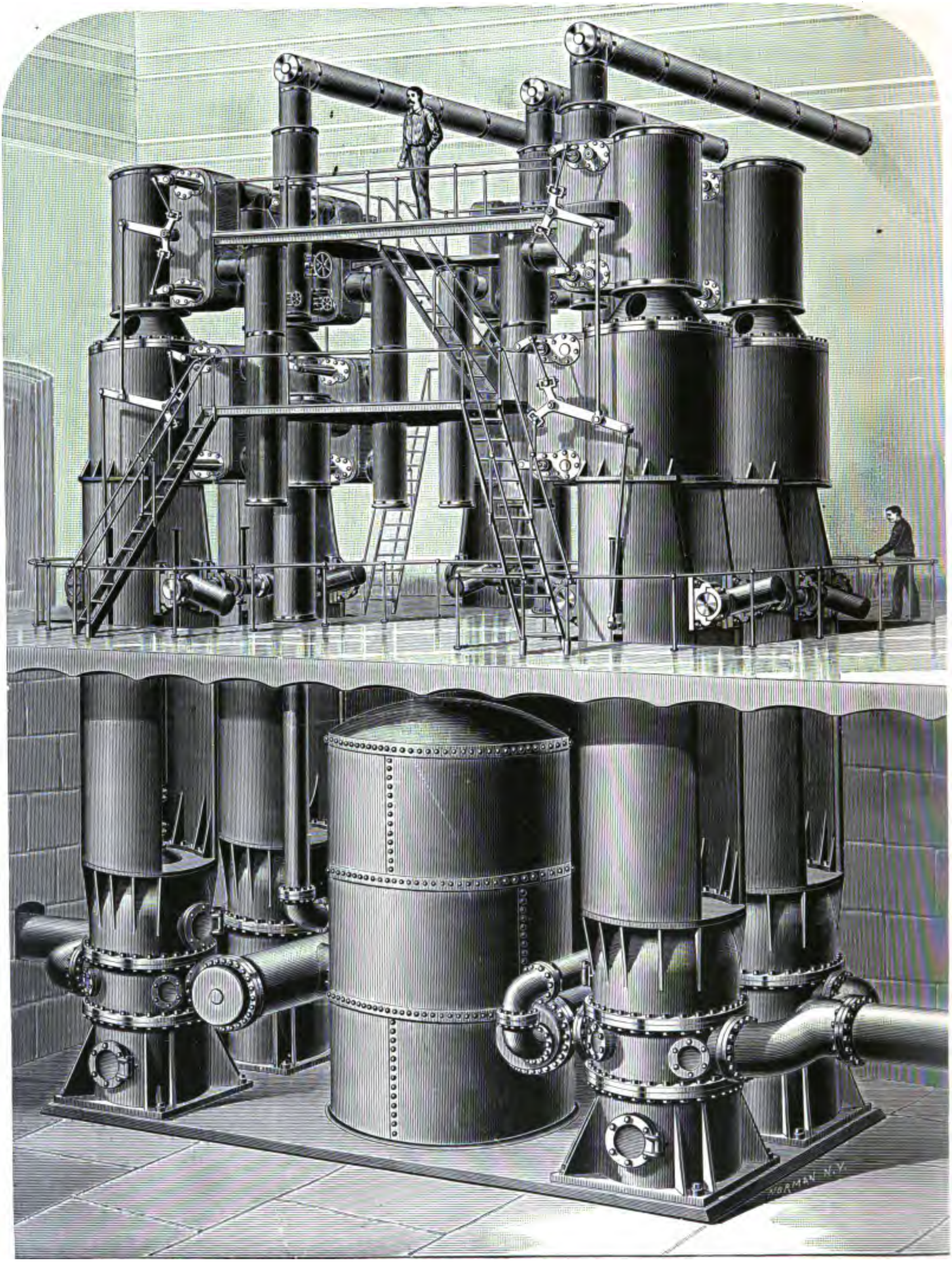
CONCLUDING REMARKS.

Throughout the trial the engine worked steadily, and was under complete control, notwithstanding the variations of pressure and the continual oscillation of the water column due to the intermittent action of the Cornish engines, with which this engine co-operated. The effect of this was noticeable in a slight periodic oscillation of the middle points of successive strokes, about the true mean mid-stroke point. The general diagram shows the variations in pressure which took place. In the early morning hours, especially when intermittent supplies were being given, these changes in pressure took place without the slightest warning. In conclusion, I have to thank Mr. Hervey, the engineer to the West Middlesex Water Company, and his assistant, Mr. Rutter, for the assistance which they afforded me in conducting this trial, and in recording the various observations. I have also to record my obligation to Mr. Mair-Rumley and to Mr. C. J. Hobbs, of Messrs. J. Simpson & Co., for the excellent arrangements which they made for the purpose of this trial." This report is dated 19th December, 1890, and signed Osbert Chadwick.

For the purpose of comparison, we append a table giving the results of trials made with Worthington and also beam engines, and it will be seen that on the same lift the efficiency of the Worthington is higher than that of the beams. This is naturally so, as the absence of the friction of the crank shaft, pins, connecting-rod, and parallel motion, etc., cannot do otherwise than raise the efficiency; and we see from the table it comes out about 6 per cent. greater. We understand Messrs. James Simpson & Co. are making some triple-expansion compensated Worthington engines, and we look forward with interest to the results that will be obtained with them.

RESULTS OF TRIALS OF PUMPING ENGINE MADE BY MESSRS. JAMES SIMPSON AND CO., LIMITED.

	Lambeth Water Works, Ditton.	West Middlesex Water Works, Hammersmith.		West Middlesex Water Works, Hampton.		New River Water Works, Stoke Newington.	West Middlesex Water Work, Hammersmith.	Oxford Water Works.
Type of engine	Receiver beam.	Woolf beam.		Worthington.		Worthington.	Worthington.	Worthington.
Engineer who conducted the trial	Mr. E. A. Cowper, M. I. C. E.	Mr. Thos. Hack, M. I. C. E.		Professor Unwin, F. R. S.		Mr. E. A. Cowper, M. I. C. E.	Mr. Osbert Chadwick.	Mr. W. H. White.
Head on pump, in feet	35.0	187.7	187.2	53 7	60.6	148.5	190.35	185.86
Efficiency per cent	77.4	83.8	85.0	84.9	84.3	91.5	87	92.3
Feed water per actual water horse-power, jackets being in circulation—pounds per hour	18 4	17 3	17.3	17.6	17.9	16.13	16.2	18.14 Including jackets.
Pounds of water evaporated per pound of coal (ash and clinker included) on trial from the feed temperature	8.347 The jackets being in circulation.	9.44 The jackets being in circulation.	9.37	9.914 Including jackets.	9.914	7.74 The jackets being in circulation.	8.01 Jackets circulating.	10.2 Including jackets.
Duty in pounds of water raised 1 foot high per 112 pounds of coal, including ash and clinker, which are not deducted, and assuming the same evaporation as was found by Professor Unwin, the full pump displacement being taken in all cases	112,626,200	118,094,000	117,650,000	111,500,000	111,500,000	121,032,000	123,300,000	128,200,000



THREE WORTHINGTON HIGH-DUTY PUMPING ENGINES

VERTICAL PATTERN.

MEMPHIS, TENN.

30,000,000 GALLONS CAPACITY.

THREE ENGINES.

FINAL REPORT OF J. J. DE KINDER, A. J. CALDWELL AND CHAS. B. BRUSH, ENGINEERS, SELECTED TO TEST THE WORTHINGTON ENGINES AND HEINE BOILERS BUILT FOR THE ARTESIAN WATER COMPANY OF MEMPHIS, TENN., JANUARY 10TH TO 14TH, 1891.

To the Artesian Water Company, Memphis, Tenn., and Henry R. Worthington, 86 *and* 88 *Liberty Street, New York City.*

GENTLEMEN—On January 27th, 1891, we submitted to you a preliminary report in relation to our test of the Worthington Engines and Heine Boilers built for the Artesian Water Company of Memphis, Tenn.. We now submit our final report in detail.

REQUIREMENTS OF CONTRACT.

The contract between the Artesian Water Company and Henry R. Worthington, dated August 18th, 1888, provides that:

"The capacity of each pumping machine should not be less " than ten million U. S. gallons for each twenty-four hours against " a water pressure of 250 feet of water to the square inch. Pump " plunger speed for this capacity not to exceed 150 feet per minute.

"The steam cylinders must be so proportioned as to do the " work called for with a steam boiler pressure of 100 pounds to the " square inch.

"There will be six (6) steam boilers required; these boilers " shall be of such size that any two of them, with economical firing, " will furnish ample steam for working one pumping machine when " running at the capacity and against the water pressure called for; " the working steam pressure in boilers to be 110 pounds to the " square inch.

"The boilers must, at the time of the test, develop an evaporative efficiency of not less than ten pounds of water from and at "two hundred and twelve degrees Fahr. for each pound of coal "consumed. The best Youghiogheny, second pool, Pittsburgh "coal to be used as the standard, and the steam pressure in the "boilers to be carried at about one hundred and ten pounds per "square inch.

"The pumping engine must, at the time of the trial herein provided for, develop a duty of one hundred and five million foot-pounds for each one thousand pounds of steam at about one hundred and ten (110) pounds pressure to the square inch in the "boilers, the steam to be taken, as regards saturation and entrainment of water, as delivered to the steam cylinders by the boilers "and fixtures furnished by the contractor.

"The construction and arrangement of the working parts of "the plant shall be such as to permit of ready access to, and easy "removal of, all such parts, for the purpose of repairs and inspection.

"Each engine shall be subject to a joint trial for capacity and "duty, said trial to be not less than twenty-four (24) hours in each "case. The test shall be made under the direction of a committee "of experts, consisting of three persons, each party to the contract "to select one of these experts, and the two thus selected to appoint "a third member, the method of conducting the trial of the boilers "and of the pumping machinery to be as follows:

"*First.*—The committee of experts may select any pair of boilers to test that they may desire, steam will be raised in the boilers to the proper working pressure, about one hundred and ten "(110) pounds per square inch, with clean fires of usual thickness, "from this time the quantity and temperature of the water put into "the boilers shall be accurately measured, and the run made for not "less than twenty-four (24) hours. All the coal used during the trial "shall be charged to the boilers, without deduction of any kind; at "the close of the trial the fire must be in as good condition, and "the water in the boilers at the same height, and the steam at the "same pressure as at the beginning of the trial.

"*Second.*—The test of the pumping machines is to be measured "in foot-pounds of work for each one thousand (1,000) pounds of "steam, as delivered to the steam cylinders by the boilers and fixtures furnished; the steam to be measured by the quantity of "water delivered to the boilers during the time of trial. The mode "of measuring the amount of work done in foot-pounds by the engine during the trial shall be determined by the Board of Experts.

" At the time of the test the machine shall deliver not less than ten " million (10,000,000) U. S. gallons of water per twenty-four " hours.

" The contractor shall guarantee the machinery and appurte- " nances for two years after the acceptance of the same (which shall " be after a satisfactory duty and capacity test) from all loss or dam- " age arising from defects in materials used, or from faulty work- " manship or design in any of the parts thereof.

" It is to be distinctly understood that the pumping plant called " for by these specifications is to be first-class in every particular, " of design, material, and workmanship, and thoroughly safe as re- " gards strength of the various parts, the desire being to secure the " best class of pumping machinery, boilers and appurtenances, such " machinery as will give an economical duty in every-day work, " and that will be thorough and efficient."

The agreement between Henry R. Worthington and the Heine Safety Boiler Company, dated October 17th, 1888, requires:

" The Heine Safety Boiler Company to guarantee that each of " the boilers shall be capable of evaporating 4,700 pounds of water " per hour into dry steam under a gauge pressure of 110 pounds " per square inch, and further that each of said boilers shall be ca- " pable of evaporating 4,700 pounds of water per hour from, and " at 212 degrees Fahr., at the rate of 10 pounds of water per each " pound of best second pool Youghiogheny coal burnt on the grates, " it being understood that no allowance is to be made from this test " for ashes, cinders and similar products; it is also agreed that " these tests shall conform to the requirements and stipulations of " the specifications of the Artesian Water Company, under which " these boilers are constructed."

In accordance with your instructions, we met at the office of the Artesian Water Company on the morning of January 10th, and after consultation with its president, Judge T. J. Latham, we proceeded to the pumping station to make the necessary preparations for proceeding with the test.

The plant furnished by Henry R. Worthington consists of three vertical direct acting pumping engines, contract capacity of each ten million gallons per day; also six Heine boilers, any two of which boilers being sufficient to run either engine at its full capacity, together with all necessary connections and appurtenances.

We found one of the three Worthington engines, No. 481, and two of the six boilers, No. 2 and No. 4, at work on the direct pressure supply connected with a stand-pipe located about two miles from the pumping station.

The test of engine No. 479 and boilers No. 1 and No. 3 was made from 3 P. M. January 12th to 3 P. M. January 13th. Before and after this test the other two engines, No. 480 and No. 481, and the other four boilers, No. 2, No. 4, No. 5 and No. 6, were examined and found to work satisfactorily.

All the water used, whether by boilers, separators, steam jackets or heaters, was charged against the plant. The water was first weighed in tanks and afterwards measured by meter. The temperatures were taken and proper allowance made therefor. The coal was weighed as used, and the time of the different weighings carefully noted. Records were taken simultaneously at fifteen minute intervals, of the length and number of strokes, the steam and water pressures, the temperatures, the draft and the meter.

Seventy-three indicator cards were taken during the test from the high pressure, low pressure and water cylinders.

The steam was examined, from time to time, by means of calorimeter tests.

All gauges and thermometers were tested with instruments that had been standardized by Schaeffer and Budenberg, of New York City, and by the Stevens Institute, of Hoboken, N. J. All scales used in connection with the feed water, coal and calorimeter were tested both before and after trial with standard weights.

The results of the test, as given in this report, are those obtained after the necessary corrections have been applied to all the readings as determined by the standard instruments. The tables and constants which have been used were obtained from the 1890 edition of D. K. Clark's Manual. In obtaining the accumulative results, as shown on Exhibits No. 1 and No. 2, proper allowances were made in the consumption of the feed water due to the difference of level of the water in the boilers for the period indicated.

The thermometer used for ascertaining the temperature of the steam was accidentally broken very shortly after the beginning of the test, and records could not be obtained. In order, however, to obtain the data for comparison, we have completed this record by giving to the steam a temperature due to its pressure as determined from the steam tables in Clark's Manual.

DESCRIPTION OF ENGINES.

The engines are compound duplex condensing high duty, of the vertical type. The steam cylinders are jacketed on both sides and ends. Each engine has a separator in steam-pipe directly over main throttle valve. The exhaust from high pressure cylinders passes through a re-heater to the low pressure cylinders.

The water from the separators, re-heaters, and jacket drains is collected in a tank and returned to the boilers by a small independent pump. During the test the water from jackets, re-heaters and separator was measured, but not returned to boilers supplying steam for the engine being tested. The condenser is of the surface condensing type located in the suction pipe of the engine, with an independent air pump for removing the vapor and condensed steam. The steam valves of both the high and low pressure cylinders are of the semi-rotating cylindrical form, and are operated as usual in the duplex form of engine. The cut-off valves are similar to the steam-valves and are operated by direct link connections easily adjusted, each side of the engine driving its own cut-off valves.

The pumps are double acting, having 28 suction and 28 delivery valves for each plunger displacement. The valves are of the disc pattern, 4¾ inches diameter, and have a lift of ⅝ of an inch.

Each side of the engine is provided with a balancing device by which the loads on the up and down strokes are equalized. This device consists of a plunger attached directly to the engine cross-head, the plunger working in a cylinder filled with water. This cylinder is connected by pipes with an air tank which is kept filled with air under pressure.

DESCRIPTION OF BOILERS.

The boilers are of the ordinary water tube type, built under the patents of the Heine Safety Boiler Company. They have a long cylindrical drum set with the rear end 18 inches lower than the front, from which, at either end, are hung two deep and broad water legs strengthened by hollow stay bolts and gathered together and narrowed at the top, where they are flanged on to the cylindrical drum. The boiler tubes of ordinary type are expanded into the inner sheets of the water legs, and connect the one water leg with the other. Opposite the opening of each of the water tubes on the outside sheet of the water leg are placed hand-hole plates which can be removed for inspecting and cleaning tubes. The furnaces, which are of the ordinary type, and the boilers are inclosed in a setting of brick. The gases pass from the grate backwards over the bridge wall to the rear of the boiler, where they are deflected by means of tiles placed among the tubes, and carried upward again around the tubes to the front end of the boiler. Here they are again deflected and carried back along the surface of the cylindrical drum, passing out into the flue at the rear end of the setting. The water in the boiler circulates from the front of the cylindrical drum backwards to the rear of the boiler where it passes downward through the rear water leg, there turning and coming forward through the tubes and rising in the front water leg. A deflection plate is placed over the opening of the front water leg to prevent the possibility of water splashing upwards and being caught by the current of steam passing to the engine. The boilers are furnished with the ordinary fittings common to all well appointed boiler plants, and are supplied with feed water by means of two independent feed pumps, either of which has sufficient capacity to supply the plant.

DESCRIPTION OF WATER SUPPLY AND SURROUNDINGS.

Memphis is located on the east side of the Mississippi River, about 350 miles north of the Gulf of Mexico. Its water supply was formerly taken from the Wolf River, a tributary of the Mississippi, just north of Memphis. Subsequent investigations made by R. C. Graves, Esq., now Vice-President of the Water Company, developed the fact that a large subterranean supply of water existed under the city. As the Wolf River supply was not satisfactory, Mr. T. J. Whitman, civil engineer, was retained as consulting engineer, and the present works were built under his direction. The work was practically completed before the death of Mr. Whitman in 1890.

The water is pumped from the large brick well, as shown on the accompanying map marked Exhibit 3. This brick well is supplied with water through a brick tunnel five feet in diameter, from forty-one tube wells. The connection between the tunnel and the brick well is six feet in diameter. Forty-two tube wells are shown on Exhibit No. 3, but well No. 1 was not in use. Ten of these wells as shown are six inches in diameter, and thirty-one of the wells are eight inches in diameter. These wells range from 260 to 480 feet in depth, the majority ranging from 350 to 400 feet. They extend into the water-bearing stratum from 80 to 130 feet. The above dimensions include the strainer, which is one-half of an inch in diameter less than the diameter of the casings. The area of the perforations of the strainer is ten times the area of the cross section of the tube.

From the pump-house the distance to the most westerly well is about 850 feet, to the most southwesterly well 1,250 feet, to the most easterly well 600 feet, and to the most northwesterly well 410 feet. These distances were measured in a direct line.

The water-bearing stratum is about 700 feet in depth. It is composed of white sand of various sizes, but most of this sand will pass through a screen of 2,500 meshes per square inch. Scattered through this sand are found beds of clay from one to three feet in thickness.

At Greenville, 125 miles south of Memphis, this bed of sand is encountered about fifty feet lower than at Memphis; at Vicksburg, 200 miles south of Memphis, a boring 800 feet in depth failed to strike this sand stratum. It is believed that this sand stratum curves to the east after passing Greenville, and continues to the Atlantic Ocean at some point near Savannah, a distance from Memphis of about 750 miles.

This water-bearing stratum of sand 700 feet thick is found underlying the Memphis region. The top of the stratum at Memphis is thirty-five feet above sea level and 150 feet below low water in the Mississippi River. This sand has a stratum of impervious clay 150 feet thick above it, which acts as a cover to retain the water and exclude all impurities. The water obtained from this sand bed has no connection with the surface about Memphis, nor with the water of the Mississippi and Wolf Rivers. The water rises from this sand to a level seven feet above high water in the Mississippi. The drainage area which supplies the water to this sand stratum is probably a belt of hilly country from fifty to eighty miles easterly from Memphis. From this belt the slope is gradually westerly and northwesterly to the Mississippi River. The beds of the largest streams of this belt are from 100 to 200 feet above high water at Memphis, while the smaller streams are still higher.

During the test Engine No. 479 pumped against a partially closed gate on the delivery main, and the water pumped by this engine was allowed to flow over a weir. During the test, also, Engine No. 481 pumped water in the distribution pipes of the city. Both engines, of course, drew their supply from the same brick well.

While the quantity of water pumped by both engines during the 24-hour test amounted to nearly twenty million gallons, the net amount drawn from the well probably did not exceed eight million gallons. The principal amount of the water that passed over the weir was returned to the brick well, and only a small portion of the surplus passed off through a sewer.

The water in returning from the weir to the brick well carried air down with it. Some of this air was caught by the current passing into the suction pipe of the engine and drawn up into the pump cylinders, the effect of which is shown by the water cylinder card. All other connections between the portion of the plant tested and the balance of the plant were entirely cut off.

The average elevation of water in the brick well during the test was 170.17 feet above sea level. The average elevation of the water in Tube Well No. 30, distant 128 feet from the brick well, was 170.78 feet; in Tube Well No. 18, distant 978 feet, it was 171.07 feet, and in Tube Well No. 7, distant 1,643 feet, the average elevation of the water was 171.63 feet. These distances were measured along the pipe line.

Exhibit No. 3 shows the elevations of water in the brick well each hour during twenty-four hours, while the engine was pumping therefrom at different rates of speed, ranging 6,927,000 to 10,980,000 gallons per day. This data was obtained by Chief Engineer E. L. Cooley on December 9, 1890, about one month prior to our test.

SUMMARY OF ELEVATIONS AT MEMPHIS, IN FEET ABOVE SEA LEVEL.

Extreme high water, Mississippi River	218.70
Extreme low water, Mississippi River	182.20
Water in wells when at rest	225.70
Top of wall of well	221.06
Floor of dry well	173.5
Floor of wet well	143.5
Bottom of suction pipe	150.5
Bottom of tunnel 6 feet diameter at wet well	146.
Top iron cover of wet well	174.34
Zero of gauge for float	175.57
Centre of plunger	178.84
Zero of tubes for vacuum on suction	175.67
Floor of engine room	220.5
Top of stand-pipe of engine	231.38
Centre vacuum gauge on Engine No. 479	236.02
Centre steam gauge on Engine No. 479	236.02
Bottom of tap for steam gauge	241.07
Crest of weir	223.81
Point on hook gauge at zero	224.6825
Fire line on boilers bottom of arch	228.65
Centre of water gauge of Engine No. 479	236.02
Elevation of top of stand-pipe at Emerson and Talbot streets,	443.50
Elevation of bottom of stand-pipe at Emerson and Talbot streets	283.50

DUTY TRIAL OF ENGINE.

DATA IN RELATION TO TEST OF WORTHINGTON ENGINE NO. 479 AND BOILERS NO. 1 AND NO. 3, DURING TWENTY-FOUR HOURS, AT MEMPHIS, TENN., JANUARY 12TH AND 13TH, 1891:

DIMENSIONS.

Number of steam cylinders, - - - - -	4
Diameter of steam cylinders, low pressure, - -	60 ins.
Diameter of steam cylinders, high pressure, -	30 ins.
Diameter of piston-rods of steam cylinders, low pressure, - - - - - - - - -	5.75 ins.
Diameter of piston-rods of steam cylinders, high pressure, - - - - - - - - -	5.625 ins.
Normal stroke of steam pistons, - - - - -	4 ft.
Number of water-plungers, - - - - - -	2
Diameter of plungers, - - - - - -	27 ins.
Diameter of plunger-rods of water cylinders, -	5.25 ins.
Nominal stroke of plungers, - - - - -	4 ft.
Net area of plunger, - - - - - - - -	561.73 sq. ins.
Net area high-pressure steam piston, - - - -	694.43 "
Net area low-pressure steam piston, - - - -	2,802.02 "
Average length of stroke of steam pistons during trial, - - - - - - - - - -	4.1625 ft.
Average length of stroke of plungers during trial,	4.1625 "

FEED-WATER.

Weight of water supplied to boilers, tank measurement,	228,247 lbs.
Weight of water supplied to boilers, meter measurement,	232,024 "
Excess of meter measurement over tank measurement,	1.66 per ct.

AVERAGE PRESSURES.

Pressure indicated by gauge on boiler, - - - -	110.06 lbs.
Pressure indicated by gauge of steam at engine, - -	105.16 "
Pressure indicated by gauge on force main, - -	95.67 "
Vacuum indicated by gauge on suction main, - -	4.85 ins.
Pressure corresponding to vacuum given in preceding line, - - - - - - - - - -	2.38 lbs.
Vertical distance between the centres of the two gauges,	60.35 ft.
Pressure equivalent to distance between the two gauges,	26.13 lbs.
Net load on plungers per square inch - - -	124.18 "

MISCELLANEOUS DATA.

Duration of trial,	24 hours.
Total number of strokes as per counter during trial,	23,057
Percentage of moisture in steam supplied to engine,	Dry.
Mean effective pressure measured from 61 diagrams taken from steam cylinders:	
High-pressure cylinder,	48.82 lbs.
Low-pressure cylinder,	14.666 “

PRINCIPAL RESULTS.

Net work done in the 24-hour test,	26,779,100,000 ft.-lbs.
Duty per 1,000 pounds feed-water,	117,325,000 “
Capacity as calculated from plunger displacement,	11,202,000 gals.

ADDITIONAL RESULTS.

Average number of counter-strokes per minute	16
Average piston speed per minute,	133.3 ft.
Average indicated H. P. developed by the steam cylinders of the engine,	605.88 H. P.
H. P. calculated from work done,	563.5 “
Efficiency of engine,	93 per ct.
Feed-water consumed by the plant per hour,	9,510 lbs.
Dry coal actually burned per I. H. P. per hour,	1.74 “
Pounds of water evaporated per I. H. P. per hour from a temperature of feed-water of 153.26° Fahr. and at steam pressure of 110.06 pounds,	15.70 “

DATA AND RESULTS OF BOILER TEST.

DIMENSIONS AND PROPORTIONS OF EACH BOILER AND CHIMNEY.

Grate surface, 5.6 feet wide, 4 feet long, area,	22.40 sq. ft.
Air space included in grate surface,	11.20 “
Clear head over bars, front,	2.62 ft.
Clear head over bars, rear,	2.30 “
Depth under bars,	2.50 “
Pitch of bars,	None.
Water-heating surface,	1,020 sq. ft.
Superheating surface,	None.
Ratio of water-heating surface to grate surface,	45.54
Height of brick chimney,	134.50 ft.

Diameter of chimney flue,	5.50 ft.
Area of chimney flue,	23.76 sq. ft.
Ratio of grate surface to area of chimney flue,	0.94

AVERAGE PRESSURES.

Steam pressure in boiler by gauge,	110.06 lbs.
Atmospheric pressure by barometer,	29.97 ins.
Force of draught in inches of water,	0.80 "

AVERAGE TEMPERATURES.

Of water in pump well,	67° Fahr.
Of steam,	334.6° "
Of escaping gases,	493.2° "
Of feed-water,	153.26° "
Of boiler room,	73.48° "
Of engine room,	74.14° "
Of outside air,	33.75° "

FUEL.

Total amount of coal consumed,	26,310 lbs.
Moisture in coal,	1,013 "
Dry coal consumed,	25,297 "
Total refuse, dry,	1,236 "
Refuse,	4.89 per ct.
Total combustible,	24,061 lbs.
Dry coal consumed per hour,	1,054 "

RESULTS OF CALORIMETER TEST.

Quality of steam,	Dry.

WATER.

Total weight of water pumped into boiler and actually evaporated, tank measurement,	228,247 lbs.
Equivalent water evaporated into dry steam from and at 212° Fahr.,	251,660 "
Equivalent water evaporated into dry steam from and at 212° Fahr. per hour,	10,486 "

ECONOMIC EVAPORATION.

Water actually evaporated per pound of dry coal from actual pressure and temperature,	9.02 lbs.
Equivalent water evaporated per pound of dry coal from and at 212° Fahr.,	9.94 "
Equivalent water evaporated per pound of combustible from and at 212° Fahr.,	10.459 "

Number of pounds of coal required to supply one million British Thermal units, as determined by analysis, - - - - - -	77.30 lbs.
Number of pounds of coal required to supply one million B. T. U., as determined from water evaporated, - - - - - -	104.05 lbs.
Efficiency, - - - - - - - -	74.3 per ct.

RATE OF COMBUSTION.

Dry coal actually burned per square foot of grate surface per hour, - - - - -	23.53 lbs

RATE OF EVAPORATION.

Water evaporated from and at 212° Fahr. per square foot of heating surface per hour, -	5.14 lbs.

THE CALCULATIONS FROM WHICH THE PRINCIPAL RESULTS WERE OBTAINED, AS APPEAR IN THE FOREGOING TABLES, ARE AS FOLLOWS:

Equivalent evaporation from 212° Fahr.,

$$228{,}247 \times \frac{1{,}065.09}{966} = 251{,}660 \text{ pounds or water.}$$

Efficiency of boiler,

$$\frac{77.30}{104.05} = 74.3 \text{ per cent.}$$

Duty,

$$\frac{23{,}057 \times 8.325 \times 2 \times 124.18 \times 561.73 \times 1{,}000}{228{,}247} = 117{,}325{,}000 \text{ ft.-lbs.}$$

Net work done,

$$23{,}057 \times 8.325 \times 2 \times 124.18 \times 561.73 = 26{,}779{,}100{,}000 \text{ ft.-lbs.}$$

Capacity,

$$\frac{8.325 \times 23{,}057 \times 2 \times 561.73 \times 7.48}{144} = 11{,}202{,}000 \text{ U. S. gallons in}$$

24 hrs.

Indicated H. P. developed,

$$\frac{(14.666 \times 2{,}802.02 + 48.82 \times 694.43) \times 8.325 \times 23{,}057 \times 2}{33{,}000 \times 24 \times 60} = 605.88$$

H. P. of water cylinders (from net work done),

$$\frac{26{,}779{,}100{,}000}{33{,}000 \times 24 \times 60} = 563.5$$

Efficiency of engine,

$$\frac{563.5}{605.88} = 93 \text{ per cent.}$$

When the term "counter-stroke" is used in this report, it indicates four displacements or two double strokes.

In connection with the performance of the boilers, it should be stated that the quality of the coal furnished was below the standard required by the terms of the contract. Calorific analyses of best Youghiogheny, second pool, Pittsburgh coal range from 13,600 to 13,800 British thermal units, while an analysis made by the St. Louis Sampling and Testing Works, January 17th, 1891, of a fair sample of the coal used, selected, boxed and sealed under our direction, shows only 12,936 British thermal units. Hence the quality of the coal used was 5½ per cent. below the requirements of the contract. This deficiency has not been considered in any of the calculations or results given in this report. It is only referred to now to explain why the average evaporation of the boilers from and at 212 degrees of 9.94 pounds of water for each pound of coal was accepted, instead of 10 pounds, as required by the contract.

Increasing the actual result obtained by 5½ per cent., the amount which the coal used fell below the standard called for by the contract, the evaporation becomes nearly 10½ pounds of water per pound of coal.

The coal was brought from the mines in open boats during a fall of rain and snow. It was immediately used as delivered from the boats. The moisture in the coal was determined by placing a quantity in a box and letting it stand over the boilers twenty-four hours and noting the difference in weight.

The steampipe connecting the boilers and engines is about 150 feet long. The steam gauge was so located that its centre was about 5 feet below the point where the connection with the steam gauge entered the main steampipe. These two facts, the length of the steampipe and the position of the gauge, together with the bends and angles in the pipe, account for the difference in pressure of about five pounds between the steam pressures of the boilers and at the engines.

We have shown the amount of feed water consumed by the plant per hour instead of showing the amount of feed water consumed by the engine per hour, as the auxiliary pumps are a part of the plant, and in the economical results the steam which they consume should be charged to the plant.

The water of condensation from the separators, steam jackets and heaters was not returned to the boilers from which steam was obtained for running the engine under test. This was, however, very carefully measured by weight and was found to be 9.48 per cent. of the total feed water as determined by the tank measurement.

DESCRIPTION OF ILLUSTRATIONS.

The charts annexed show graphically the record of the principal data obtained during the test and the deductions therefrom.

Exhibit No. 1 shows the varying pressure of the water in pounds per square inch on the force main just beyond the pumps; the varying boiler pressure and the variation in the number of strokes of the engine, also the averages of these three records.

There is also shown on Exhibit No. 1 the duty in million foot-pounds per 1,000 pounds of water evaporated, as prescribed in the contract, and we have also shown, as forming a very interesting part of these records, the duty in 1,000,000 foot-pounds per 100 pounds of coal burned; also the average pounds of feed water evaporated per pound of coal; also the average pounds of feed water evaporated per pound of coal from and at 212° Fahr. These four records show accumulative results at the end of each three hour period from the time of the beginning of the test.

Exhibit No. 2 shows the total number of pounds of feed water evaporated, the total pounds of equivalent feed water evaporated from and at 212° Fahr., the total number of pounds of coal burned, and the total work accomplished in million foot-pounds. All four of these records are accumulative results obtained at the end of each three-hour period.

Exhibit No. 3 shows in plan the general location of the works of the Artesian Water Company, a cross-section of the dry and wet wells, together with an elevation of the pumping machinery and a section of a boring taken to determine the strata.

Exhibit No. 3 also shows in plan and section the weir and the details of its principal features.

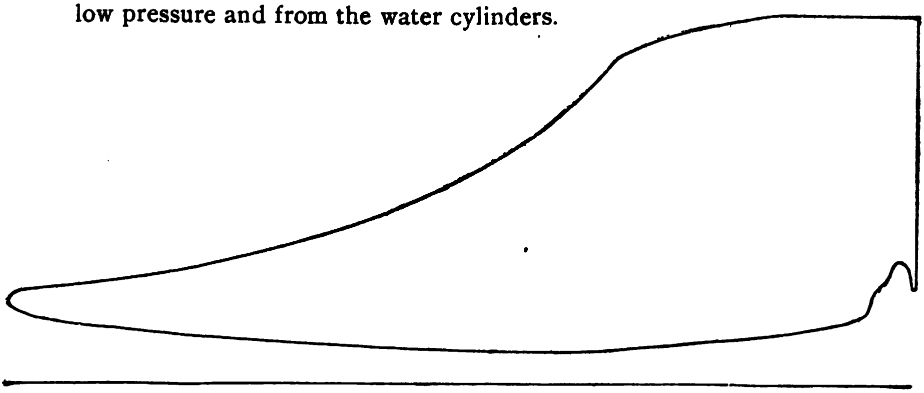
By referring to Exhibit No. 1 there will be noticed a decided change in the economy of the boilers at the end of the ninth hour from the time of the beginning of the test. Up to this hour the boilers were showing very little increase in economy, and were barely maintaining the economy reached at the end of the second period, or six hours from the start. At the end of the ninth hour a change was made in the management of the boilers, and later in the personnel of the firemen. The record thereafter shows a decided increase in the economy of the plant, and this economy continued to increase to the end of the test.

In connection with these two records, we may also consider the record of the boiler and water pressures, a study of which shows very little deviation from the average, consequently the increase in the economy of the boilers was not due to the more favorable conditions for operating the plant, but to the skill in handling the boilers.

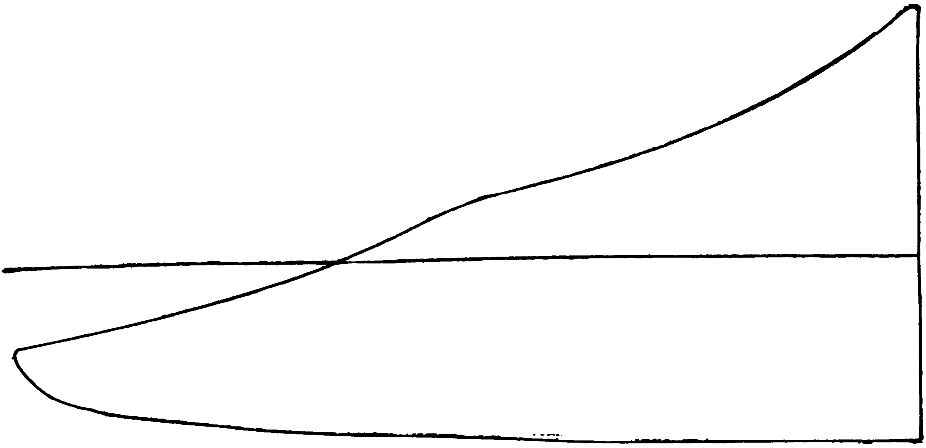
A study of the record of the duty, based on the water evaporated, shows a very uniform result, slightly increasing.

The record of foot-pounds of work is almost a straight line, and shows that the work of the engine was remarkably uniform.

The following is a fair sample of the cards from the high and low pressure and from the water cylinders.



CARD TAKEN FROM HIGH-PRESSURE CYLINDER.

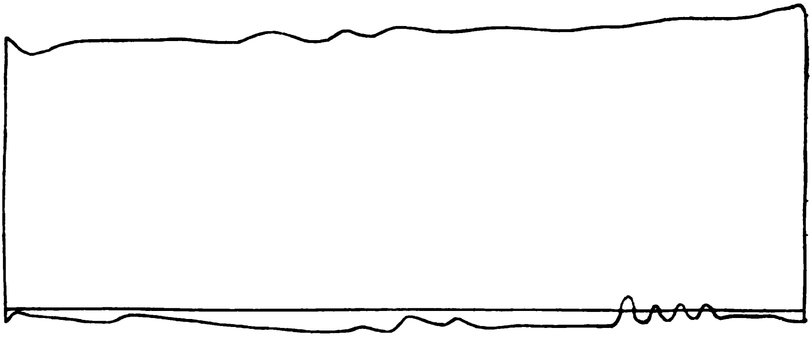


CARD TAKEN FROM LOW-PRESSURE CYLINDER.

WEIR CONSTRUCTION AND DATA.

In order to test the discharge of the pumps, the Water Company carefully constructed a masonry weir lined with Portland cement. This weir is shown in plan and detail on Exhibit No. 3. The weir chamber is about 47 feet long in the clear and 11 feet wide. The weir has two end contractions, the edge of the notch being about 3 feet from the sides of the weir. The crest is about

7 feet above the bottom of the weir chamber. The notch is 5 feet in width, and the depth of the water flowing over the weir was about one foot. The height of the water in the weir chamber was measured in a gauge chamber on the side of the weir about 10 feet back from the crest, and this gauge chamber was connected with the weir by a perforated iron pipe 4 inches in diameter about 3 feet above the bottom of the weir chamber. The water entering at one end of the weir chamber passed through three sets of stilling screens about 5 feet apart, after which the water had a clear run of about 25 feet and a free fall over the crest. The surface of the water during the test was about 18 inches below the top of the weir chamber. Planks were thrown over the top over the chamber to protect the surface of the water from the effect of the wind.



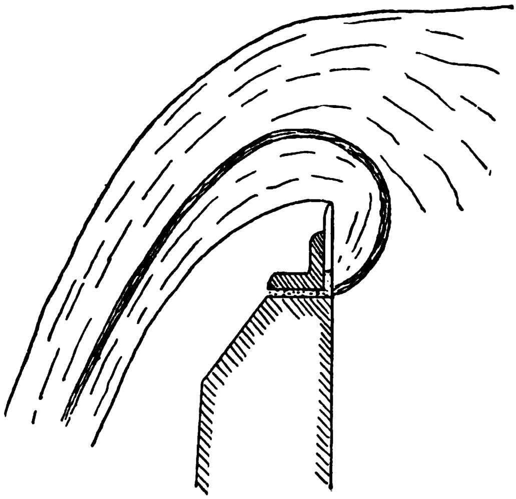
CARD TAKEN FROM PUMP CYLINDER.

From 4.16 until 5.20 P. M., on January 12th, and from 9 until 10.30 A. M., on January 13th, readings were taken at the gauge chamber with a hook gauge every 30 seconds, to determine the height of the water flowing over the weir.

During the test the following measurements were taken simultaneously in the gauge chamber and at the weir crest, in order to determine the difference in the level of the surface of the water at these points:

Height of Water over Crest of Weir, measured in Gauge Chamber.	Height of Water over Crest of Weir, measured at Weir.	Difference.
1.0395 feet.	0.9903 feet.	0.0492 feet.
1.0345 "	0.9864 "	0.0481 "
1.0335 "	0.99838 "	0.0497 "
Average, 1.0358 feet.	0.9868 feet.	0.0490 feet.

The desire of the Chief Engineer, Mr. Cooley, to have the weir test as nearly accurate as possible, caused him to delay setting the brass frame of the weir in its final position until the night before the test was made. During the night of January 11th, the Portland cement around this brass frame froze and became more or less disintegrated. During the test this cement washed out to a considerable extent, and a gray streak of fresh Portland cement from the joint just below the brass plate appeared in the water, as shown in the sketch. This streak of colored water appeared to be about one-third of the depth above the crest of the weir.



When this defect in the condition of the weir was ascertained it was too late to remedy the difficulty, and the suggestion was made to abandon the weir test. We insisted upon making this test, however, because we felt convinced that the weir, even in this slightly defective condition, would give a close approximation to the true result, and also because we were curious to know the extent to which the accuracy of the weir measurement would be affected by the defect described. We also desired to ascertain how much the results would differ, if calculated by the different formulæ of standard authorities, and also what the difference would be if the weir readings were taken at intervals of thirty seconds, or at intervals of fifteen minutes.

We selected the formulæ of J. B. Francis, Fteley and Stearns, and Hamilton Smith, Jr., for this comparison.

When the first weir readings were taken, the cement around the brass frame had been disintegrating about seventeen hours. When the second weir readings were taken the disintegration had been in process about thirty-four hours. There was no leakage of water at any time around the brass frame or from any portion of the chamber.

The following is a comparison of the discharge in cubic feet per second as obtained—from calculations based on the plunger displacement of the pumps—and from weir measurement from the formulæ given—at the two periods mentioned, the former period being seventeen hours from the beginning of the disintegration and the second period being seventeen hours later. The readings at fifteen minute intervals were taken simultaneously at the pumps, and at the weir.

RECORDS TAKEN FROM 4.15 TO 5.15 P. M., JANUARY 12TH, AT FIFTEEN MINUTE INTERVALS.

AUTHORITY.	Discharge in Cubic Feet per Second.	Percentage.
Pump Displacement	17.1700	100
WEIR FORMULA.		
Hamilton Smith, Jr.	16.8320	98.0
J. B. Francis	16.7162	97.3
Fteley & Stearns	16.6713	97.1

RECORDS TAKEN FROM 9 TO 10 A. M., JANUARY 13TH, AT FIFTEEN MINUTE INTERVALS.

AUTHORITY.	Discharge in Cubic Feet per Second.	Percentage.
Pump Displacement	17.3600	100
WEIR FORMULA.		
Hamilton Smith, Jr.	16.8126	96.8
J. B. Francis	16.6971	96.2
Fteley & Stearns	16.6526	95.9

From these tables it is apparent that the disintegration of the cement in one period of seventeen hours affected the accuracy of the result of the weir test to the extent of about 1.2 per cent. It also appears in this case that the results obtained from the formulæ given, vary about nine-tenths of one per cent.

Comparing the results obtained by taking weir readings at short and at long intervals, we found :

That between the hours of 4.15 and 5.15 P. M., on the 12th of January, the average of the weir readings taken at intervals of thirty seconds shows a *decrease* in the total discharge of *three-hundredths of one per cent.*, as compared with weir readings at fifteen minute intervals during the same period, and

That between the hours of 9 and 10 A. M. on the next day, the average of weir readings taken at intervals of thirty seconds showed an *increase* in the total discharge of *eighteen-hundredths of one per cent.*, as compared with the weir readings taken at fifteen minute intervals during the same period.

The following are the formulæ which we used for computing the discharge by Weir Measurement :

Francis—Formula :

$$Q = 3.33 \left[L - 0.1\, n \left[(H+h)^{\frac{3}{2}} - h^{\frac{3}{2}} \right]^{\frac{2}{3}} \right] \left[(H+h)^{\frac{3}{2}} - h^{\frac{3}{2}} \right]$$

Hamilton Smith, Jr.—Formula :

$$Q = \tfrac{2}{3}\, c\sqrt{2g} \times L \left(H + b\frac{Va^2}{2g}\right)^{\frac{3}{2}}$$

Fteley and Stearns—Formula :

(This is an adaptation of Fteley and Stearns' Formula to suit this case of the weir with end contractions using Francis' principle in allowing for end contractions.)

$$Q = 3.31 \left[L - 0.1\, n\, (H + Ch)\right] \times (H + Ch)^{\frac{3}{2}} + 0.007\, L$$

In all of which the same data were used. During the first period these data were :

Q = Discharge in cubic feet per second.
H = 1.0303 feet, the observed depth of water on the weir.
h = .0009 feet, the height due to the velocity of approach.
n = 2, the number of end contractions.
L = 5 feet, the length of weir.
$2g$ = 64.36.
b = Co-efficient for end contractions, in this case taken to be 1.4.
c = Co-efficient of discharge, in this case taken to be 0.601.
C = 1.98, the co-efficient in this case.

The velocity of approach, *Va*, was not measured, but was deduced from the computed discharge of the pumps and the cross section of the stream of water in the weir at about the point of hook gauge readings. This gave a mean velocity of approach of 0.24 feet per second.

CONCLUSIONS.

In conclusion, we formally submit—following the phraseology of the contract—a summary of the results of our test.

We find that the capacity of each pumping engine is 11,202,000 U. S. gallons for each twenty-four hours against a water pressure of 124.18 pounds (286.85 feet) to the square inch when running at a piston speed of 133.3 feet per minute. This is an excess in capacity of 12 per cent. over the requirements of the contract, and this result was obtained when running at a piston speed 11 per cent. less than the speed allowed by the contract. The average water pressure exceeded the contract stipulation 14 per cent.

We find that the boilers at the time of the test did develop an evaporative efficiency of ten pounds of water from and at 212° Fahr. per pound of coal consumed, taking the best Youghiogheny, second pool, Pittsburgh coal as a standard.

We find that the pumping engine at the time of the trial did develop a duty of 117,325,000 foot-pounds for each 1,000 pounds of steam, at a boiler pressure of 110.06 pounds per square inch. This duty is 11.7 per cent. in excess of the requirements of the contract. The steam was measured by the quantity of water delivered to the boilers during the time of trial. The steam was taken as regards saturation and entrainment of water as delivered to the steam cylinders by the boilers and fixtures furnished by the contractors.

We find that each boiler evaporated 5,243 pounds of water per hour into dry steam under a gauge pressure of 110.06 pounds per square inch from and at 212° Fahr. at the rate of 10 pounds of water per pound of best second pool Youghiogheny coal burnt on the grate, no allowance being made in this test for ashes, cinders and similar products. This is 11 per cent. in excess of the requirements of the contract.

This test was made with clean fires of usual thickness, the quantity and temperature of the water put into the boilers was accurately measured, and the run was made for 24 hours. All the coal used during the trial was charged to the boilers without deductions of any kind. At the close of the trial the fire was in as good condition, the water in the boilers at the same height, and the steam at the same pressure as at the beginning of the trial.

We find that the construction and arrangement of the working parts of the plant are such as to permit of ready access to and easy removal of all such parts, for the purpose of repairs and inspection.

We also find that the plant has been built in accordance with the plans and specifications, that it is first class in every particular, of design, material and workmanship, and thoroughly safe as regards strength of the various parts—that the pumping machinery, boilers and appurtenances are all first class, thorough and efficient, and such as will give an economical duty in every-day work.

The plant is an exceedingly interesting one, and in many respects it is unique and novel. We have, therefore, taken some pains to describe it fully in order that its design and construction may be clearly understood by any one who may read this report

We are under special obligations to E. L. Cooley, the present chief engineer of the works; T. T. Johnston, the former chief engineer, and James M. Safford, State Geologist of Tennessee, for valuable information and data relating to this plant.

We thank you for the courtesy, attention and ready assistance which we have received from every one we have met while engaged in this test.

Respectfully submitted.

J. J. DE KINDER,

Expert selected to represent the Artesian Water Co.

A. J. CALDWELL,

Expert selected to represent Henry R. Worthington.

CHAS. B. BRUSH,

Expert selected by both parties.

BIRMINGHAM, ALA.

5,000,000 GALLONS CAPACITY.

ONE ENGINE.

REPORT OF MESSRS. ERNEST DREYSPRING AND E. H. FOSTER.

BIRMINGHAM, ALA., April 10th, 1891.

Birmingham Water-Works Co., Birmingham, Ala.:

GENTLEMEN—The following test was made on April 6th and 7th, 1891, to determine the duty and capacity of the New Worthington High Duty Engine at the Cahaba River pumping station:

The test began at 8 o'clock P. M., April 6th, and closed at 8 o'clock P.M., April 7th, continuing without interruption for 24 hours. Readings of pressure gauges and length of stroke were taken every half-hour. The water in the boilers was brought to exactly the same level at the end as at the beginning of the test, and the water in the Hoppes Feed Water Purifier stood ¼-inch higher at the end than at beginning of test.

The feed water was measured by meter, of which 10 tests were made to establish the error of registration.

Ten calorimeter tests were made to determine the quality of the steam, but the results varied so largely that it was decided to disregard them entirely.

There were no means by which the actual slip of the pump could be measured, but from their design it would be safe to assume that such quantity would be small.

The engine operated throughout the test in a perfectly satisfactory manner, with the two following exceptions:

1st. A slight click occurred in one of the H. P. pistons, due to a slackness in the ring, which was remedied immediately after the test.

2d. Shortly after 8 o'clock A. M., the fireman allowed the water to get too high in one of the boilers, and a large quantity went over to the engine, choking up jackets and causing speed to slacken. The engine did not fully recover from this effect for about two hours, but after that time it ran as during the first half of test, and could not be criticised in any way.

DESCRIPTION OF PLANT.

Engine is of Worthington Horizontal High Duty type, and has the following dimensions:

High-pressure steam cylinders, diameter,	28¾ in.
Low " " " "	57½ "
Plunger, "	19 "
Length of contact stroke,	50 "

It is located at the bottom of a circular well, 56 feet in diameter and 25 feet deep.

Steam is furnished by a battery of horizontal tubular boilers, placed on level of ground in a separate building, 25 feet above top of engine foundation.

DUTY.

Area of 19-inch plungers,	283.52 sq. in.
" " 4¾-inch rod,	17.72 " "
" " 3½-inch "	9.62 " "
" " plungers, effective,	269.859 " "
Engine counter, 8 P. M., April 7,	985,624
" " 8 " " 6,	963,464
Number of revolutions made in 24 hours,	22,160
" strokes " " "	88,640
Average stroke of piston in feet,	4.163
Total travel of plunger in feet for 24 hours,	369,008.32
Average reading of water pressure gauge in pounds,	179.602
Average reading of suction lift (mercury col.), pounds,	5.961
Difference of level of water gauge and suction chambers in pounds,	3.671
Total pressure pumped against, pounds,	189.234
Total load on plunger, in pounds,	51,066.5
Work in foot-pounds for 24 hours,	18,843,963,373.28
Meter counter in cubic feet at 8 P. M., April 7,	15,564
Meter counter in cubic feet at 8 P. M., April 6,	12,807
Cubic feet passed through meter in 24 hours,	2,757
Average shortage of meter in 10 tests,	1.24 per ct.
Water used in testing meter (cubic feet),	60
Average temperature of feed water,	176.3 deg.

Weight of water at 176.3 deg.,	60.63
Water used in calorimeter tests (pounds),	472
Net feed water used by engine at 176.3 deg.,	165,074
Average temperature of injection water,	51.6 deg.
Equivalent feed water at temperature of injection (pounds),	147,411

Duty on contract conditions:

$$\frac{18{,}843{,}963{,}373 \times 1{,}000}{147{,}411} = 127{,}842{,}357$$

Duty guaranteed,	105,000,000
Excess,	22,842,357

CAPACITY.

Travel of plunger in feet for 24 hours,	369,008.32
Piston speed in feet per minute,	128
Plunger displacement per foot of travel, gallons,	14.018

Capacity: 369,008.32 × 14.018 = 5,172,558.629 gallons in 24 hours.

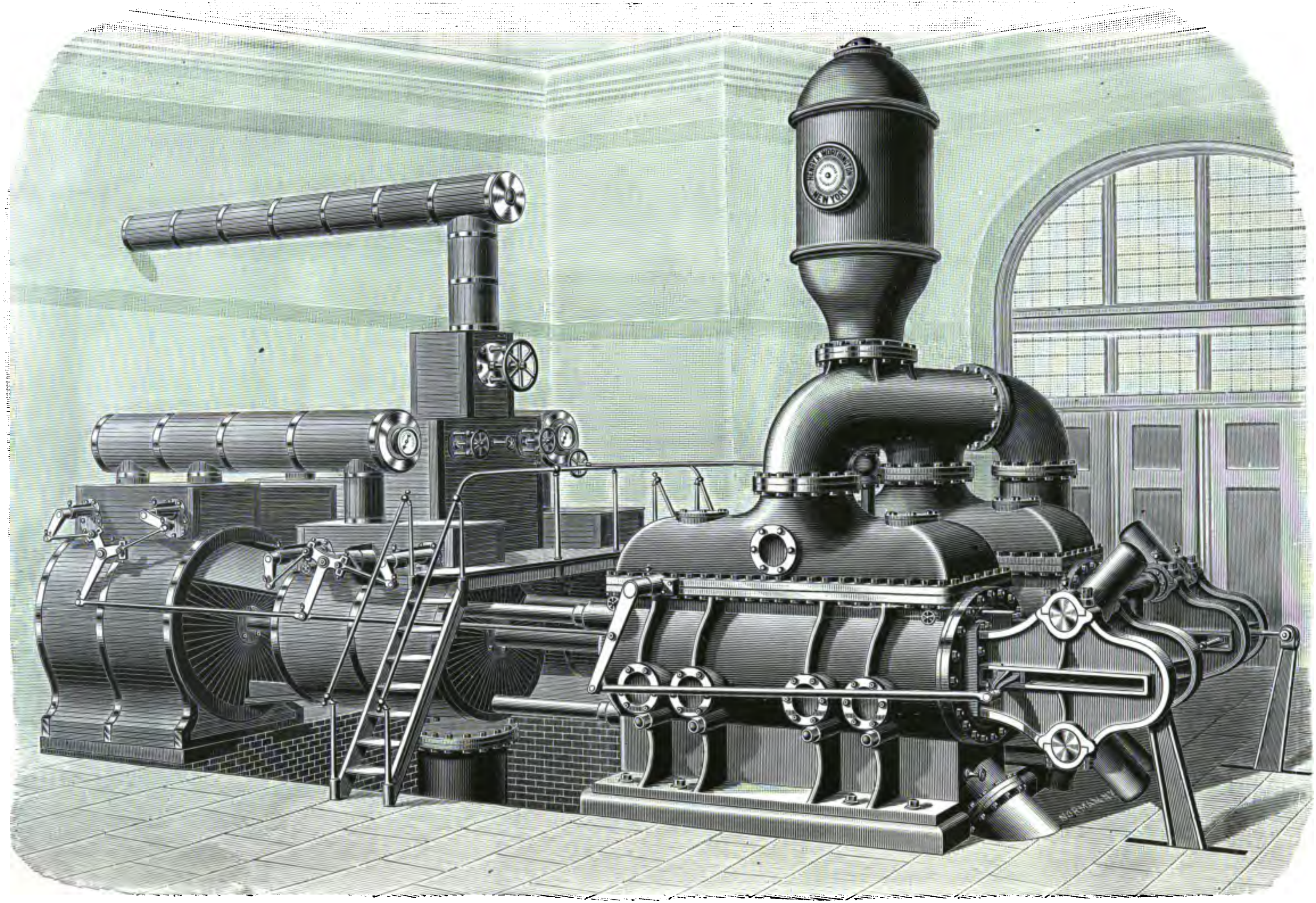
" Guaranteed, - 5,000,000

" Excess, - - 172,558 gallons.

Capacity at 125 feet piston speed—5,046,696 gallons in 24 hours. Gauges were taken to Linn Iron Works after the test and tested on their mercury gauge tester, and found to be correct.

Respectfully submitted,

ERNEST DREYSPRING,
E. H. FOSTER.



WORTHINGTON HIGH DUTY PUMPING ENGINE AT SYRACUSE, N. Y.

SYRACUSE, N. Y.

10,000,000 GALLONS CAPACITY.

ONE ENGINE

REPORT OF GEO. C. FREYER, ASST. ENGR.

ENGINEERING DEPARTMENT OF THE SOLVAY PROCESS COMPANY.

SYRACUSE, N. Y., April 16th, 1891.

EDWARD N. TRUMP, *Esq., Chief Engineer:*

DEAR SIR—The following is my report of the duty trial made on April 7th, 1891, of the Worthington High Duty Steam Pumping Engine, located at the Lake Pumping Station of the Solvay Process Company, Syracuse, N. Y., in accordance with your orders.

The contract made with Henry R. Worthington, of New York, calls for a steam pumping engine with two high pressure cylinders of 21 inches diameter, two low pressure cylinders 42 inches diameter, and two double acting water plungers of 27½ inches diameter; length of stroke to be 36 inches; speed of piston to be about 118 feet per minute, and to have a pumping capacity of 10,000,000 gallons of water in 24 hours against a total pressure, including suction, of 60 pounds per square inch, with an effective steam pressure of 120 pounds per square inch. The expansion to be at least 12 times, and its total capacity or duty was to be 100,000,000 foot-pounds of work per 1,000 pounds of steam used.

The plunger displacement system of measurement is the one I have employed in determining the duty. The slip in the pump seemed to be so little that I left it out of my calculations entirely. I took no account of the amount of work done in overcoming the friction of the water in passing through the passages and valves of the pump. Great care was taken to prevent any air from entering the pump during the trial, thereby preventing the possibility of imperfect filling. The necessary data having been obtained according to your orders, the duty of the engine and other quantities relating to the performance of the pump I have calculated therefrom.

The following are the principal data, to which I have appended my calculations, all of which is respecfully submitted.

Yours very truly,
GEO. G. FREYER,
Assistant Engineer.

[Approved] EDWARD N. TRUMP,
Chief Engineer.

DATA AND RESULT OBTAINED IN MAKING THE TEST OF THE WORTHINGTON HIGH-DUTY PUMPING ENGINE, BY GEORGE G. FREYER, ASSISTANT ENGINEER AT THE LAKE PUMPING STATION OF THE SOLVAY PROCESS COMPANY, SYRACUSE, NEW YORK.

Date of test,	April 7th, 1890.
Time of beginning,	8 o'clock A. M.
Time of ending,	6 o'clock P. M.
Duration,	10 hours.

PRINCIPAL DIMENSIONS OF ENGINE.

Diameter of each high pressure cylinder (2),	21 in.
“ “ “ low “ “ (2),	42 “
“ “ “ plunger (2),	27½ “
Length of stroke,	3.166 ft.
Diameter of high pressure piston rod,	4¼ in.
“ “ low “ “ “	4¼ “
“ “ plunger piston rod,	4¼ & 3¼ “
Mean area of high pressure cylinder,	332.175 sq. in.
“ “ “ low “ “	1,378.36 “ “
“ “ “ plunger	582.718 “ “

AVERAGE READING AND RESULT.

Average steam pressure at engine,	120.83 lbs.
“ vacuum,	28.5 in.
“ water pressure,	50.6 lbs.
“ suction gauge, 13.44 ft., =	6.58 “
Distance between gauges, 6.145 ft., =	2.67 “
Total lift,	59.85 ‘

Average length of stroke,	3.166 ft.
“ temperature of feed water at meter,	78.83° Fahr.
“ “ “ lake “	40.4 “
“ “ “ air pump discharge,	82.9 “
“ “ “ hot well,	80.1 “
“ “ “ engine room,	73.9 “
“ “ “ outside air,	37.2 “
“ height of barometer,	29.60 in.
Total number of displacements,	44,328
Average per minute,	73.88
Average piston speed, (feet per minute),	116.95 ft.
Total gallons pumped at test,	4,294,193 gals.
Rate of pumping for 24 hours,	10,198,064 “
Amount of water by meter to boilers,	40,799.25 lbs.
Water collected in drop-leg or separator,	221.82 “
Water returned from jackets,	4,313.7 “
Net water used in boilers,	44,891.13 “
Entrained water in steam, about	5 per ct.

$$\text{Duty} = \frac{582.718 \times 59.85 \times 44{,}328 \times 3.166}{44{,}891.13} \times 1{,}000 = 109{,}031{,}592 \text{ ft.-lbs.}$$

Indicated horse-power, steam end,	272.0
Indicated horse-power, water end,	254.5
Efficiency of engine,	93.5 per ct.
Feed water per hour,	4,489 lbs.
Feed water, per indicated horse-power, per hour,	16.5 “
Evaporation per pound of coal,	7.01 “
Coal consumed,	6,400 “

NATIONAL TRANSIT COMPANY,

SWARTOUT, N. Y.

PERFORMANCE OF A WORTINGTON HIGH DUTY PUMPING ENGINE OF ONE AND ONE-HALF MILLION GALLONS CAPACITY PER TWENTY-FOUR HOURS, AGAINST A HEAD EQUIVALENT TO TWO THOUSAND FEET OF WATER.*

BY J. E. DENTON, HOBOKEN, N. J.

INTRODUCTORY.

THE pumping engine under notice is in use for pumping crude petroleum over one of the eleven 30-mile sections of the Standard Oil Pipe Line connecting New York City with Olean, Pa.

It is located in a valley of the Alleghany Mountains at Swartout, about four miles from Port Jervis, New York. A bird's-eye view of the pumping station is given in Fig. 282. The general features of such a pumping station are a boiler and engine house, a telegraph office, and two receiving tanks nearly 100 feet in diameter and 30 feet in height. Oil is supplied to the tanks from another station 30 miles westward, by 6-inch pipes running across the country a few feet beneath the surface of the ground. The light streak running up the mountain at the background of Fig. 282 shows the route of the pipe. It represents the path worn by the passage of the "line walkers," who constantly patrol the line. A telegraph operator belonging to each station, measures the depth of oil in the tanks every hour of the day and night, and reports the result throughout the line, so that the utmost system prevails, and every station is under perfect control from the headquarters. Fig. 283 shows an operator in the act of measuring the depth of oil in a tank by means of a steel tape dropped to the bottom of the tank from a reel held in his hand.

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* Presented at the Providence meeting (1891) of the American Society of Mechanical Engineers, and forming part of Volume XII. of the *Transactions*.

At night, sufficient light is thrown upon the tape to enable it to be accurately read, by means of a reflector mounted upon a tower situated midway between the tanks and at such a distance that all danger from fire is avoided. Figs. 284 and 285 show general views of the pumping engine. In the foreground are the oscillating or compensating cylinders, constituting the high duty attachment which has enabled the expansive principle of using steam to be developed in the Worthington direct acting pump, so as to afford a saving of coal amounting to about 50 per cent. of the consumption involved without this attachment.

FIG. 282.

Bird's-eye view of pumping station, showing boiler and engine houses, two oil receiving tanks, 95 feet in diameter and 30 feet high, with light tower between, to enable readings of depth of oil in tanks at night, as per Fig. 283. The route of the pipe is shown by the light streak running over the mountain in the background, which represents a path worn by the passage of the "line walkers," who constantly patrol the country through which the pipe runs. Eleven such stations, about 30 miles apart, enable one and one-quarter million gallons of oil to be transported daily from Oil City, Pa., to New York City.

The pump tested represents the first application of the high duty attachment to the pipe-line service. This service is very severe, inasmuch as the pressure which the pump must exert often amounts to upward of 900 pounds per square inch, and the work is continuous throughout the day and night. By aid of the devices shown in Fig. 286 and Fig. 287, the conditions of service are maintained comparatively uniform, and consequently afford excellent opportunity for the accurate measurement of the duty of a pump under the actual conditions of practice; and the object of this paper is to present the result of a series of duty trials recently made by the writer, in the joint interest of the pipe-line management and the makers of the pump. I am indebted to the officers of the pipe line, and especially to Messrs. Pilkington and Cobb, for cordial co-operation and assistance in carrying out the experimental work.

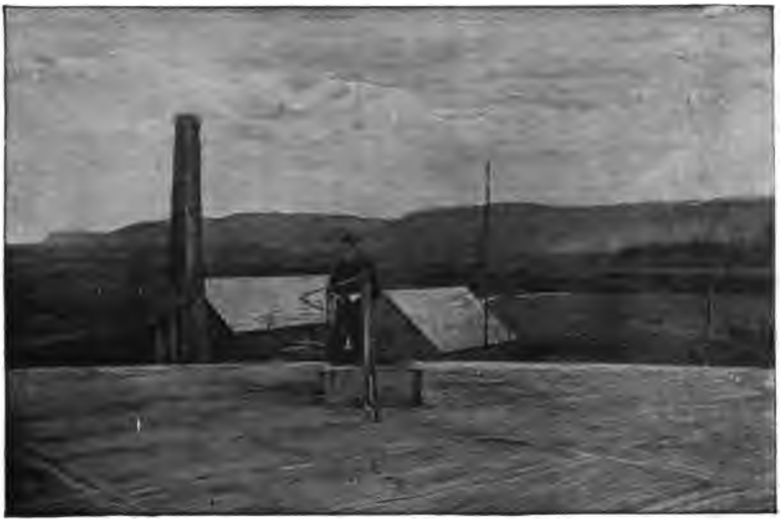


FIG. 283.

Showing station telegraph operator measuring depth of oil, in one of the receiving tanks, by means of a steel tape, dropped to the bottom, off a reel held in his hand. Measurements are made every hour, day and night, and the reading telegraphed to the adjoining stations, 30 miles distant, east and west. At night, sufficient light to read the tape is thrown upon the tape by a large reflector, located midway between the two tanks (Fig. 282), and at such a distance as to prevent any danger of ignition to the contents of the tanks.

SPECIAL PREPARATIONS FOR TESTS.

The station is provided with eight boilers, six of which are in use at one time to supply steam to operate the pumping engine, an independent steam feed pump, a small engine operating a dynamo and for heating the telegraph office and engine house. A sectional view of the engine is shown in Fig. 291. The air pumps are operated by direct connections from the cross-heads. All of the cylinders are steam-jacketed, and the steam condensed in the jackets is returned to the boilers by an automatic Pratt & Cady steam-trap. The remaining portion of feed-water is taken from the hot-well by the independent steam feed-pump, and forced through a pipe surrounded, over a portion of its length, by a larger pipe, into which the steam feed-pump and the dynamo engine and heating

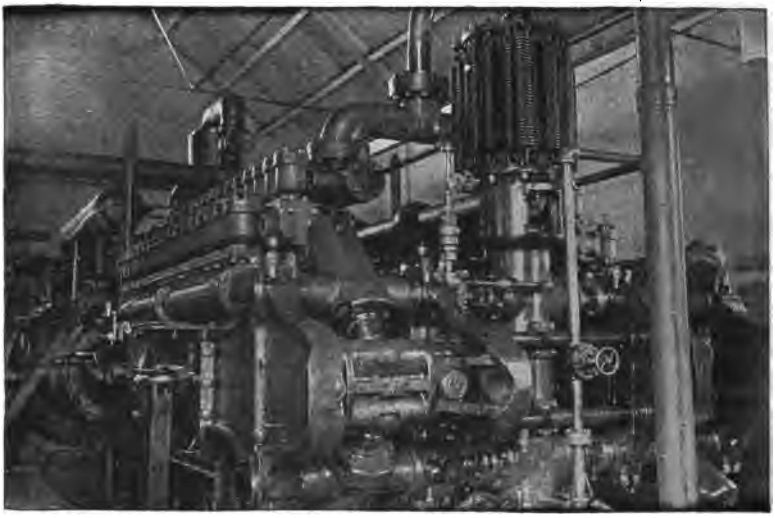


FIG. 284.

General view of pumping engines, showing 6-inch vertical delivery pipe, compensating or oscillating cylinders, and the group of springs acting as an equivalent of an air chamber. The springs resist the upward movement of a vertical plunger, 6 inches diameter, which works in and out of a cylinder connected with the delivery chamber of the pump by the 2-inch vertical pipe in front of the pump. The oscillating cylinders are discharged or filled by oil flowing through their hollow trunnions, to or from the cylinder containing this plunger, and the springs control the fluctuations of pressure which occur.

coils exhaust, thereby slightly heating the feed-water, taken from the hot-well, above the temperature in the latter.

The condensed steam from the jackets was cut off from the automatic trap and taken through a Nason dumping trap into a surface condenser, which cooled it sufficiently to allow it to be received in an open barrel, and thence pumped through a three-quarter Worthington meter, by means of a small steam pump provided for the purpose. The feed-water then all entered the boilers through the independent feed-pump, and was measured by a 2-inch Worthington meter located between this pump and the boilers. One of the six boilers was devoted to supplying steam for all purposes, except running the pumping engines, a separate steam main connecting it with the dynamo engine, the heating coils, the feed-pump, and the small pump used for forcing the jacket steam through the water. A second three-quarter meter measured the feed-water supplied to this boiler. Hence, the feed-water regis-

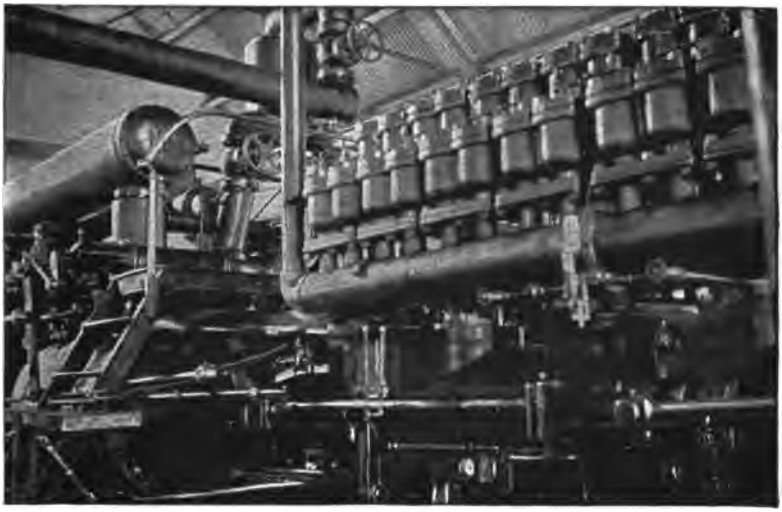


FIG. 285.

Showing lazy-tongs, indicator motion, main steam-pipe, main throttle valve, branch throttle valves to each side of engine and half-inch vertical pipe, to which the universal calorimeter was attached to measure moisture in steam supplied to engine; also 1¼-inch pipe, out of bottom of steam pipe, directly below main throttle valve, which supplies steam to all jackets, and to reheater in receivers.

tered by the 2-inch meter, less that registered by the three-quarter meter attached to the single boiler, was the amount of water evaporated into steam to operate the pumping engine under test, and included both the steam condensed in the jackets and that entering the cylinders. Coal was weighed in portions of about a ton each on the scales with which the station is provided, the latter being calibrated with an accurate steel-yard provided by the writer, and by means of which dead weight was accumulated equal to the greatest weight for which the scales were used. Indicators were

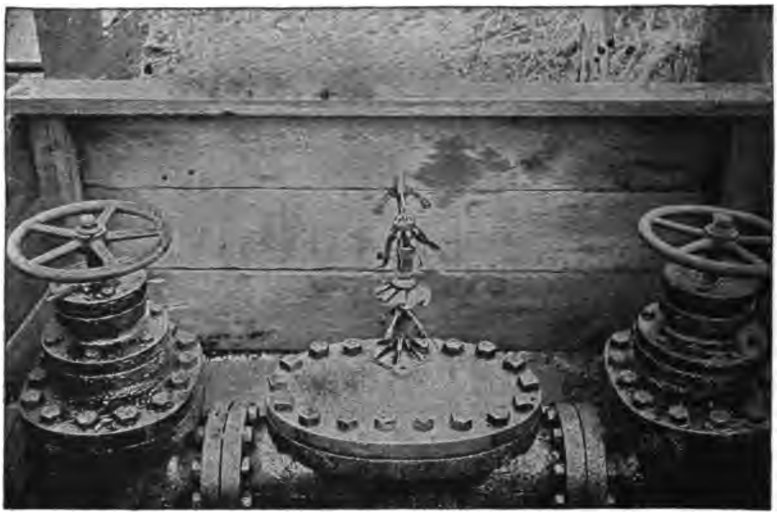


FIG. 286.

Showing the "go devil," or instrument which is allowed to travel through the oil pipe, to cleanse the interior of the latter of oil accumulations tending to obstruct the flow. The stop valves at either side are closed, and then the oblong manhole plate removed. The instrument is then laid in the pipe, the cover replaced, and the valves opened. The current of oil, which travels at the rate of about three miles per hour, pushes the "go devil" along, and also revolves it with sufficient force to cause the bent steel knives, near its upper end, to cut all residue off from the interior of the pipe, and thereby prevent undue expense of power in pumping the oil. The instrument creates an audible buzzing sound, which, together with the fact that its arrival is anticipated by telegraph notice, enables it to be caught at the next station, or as soon as it arrives at the proper depot.

fitted to all the steam and pump cylinders, and were operated by a lazy-tongs motion. A "stroke register" was provided to integrate automatically the lengths of the strokes of the pump during any interval of time.

A Barrus universal calorimeter was attached to the steam supply pipe immediately before the latter entered the main throttle valve. (See description, Fig. 285.)

Steam-gauges, carefully calibrated, were located at the boiler, at the steam-pipe, at its entrance to the engine, just beyond the main throttle, and below the branch throttles leading to either side of the engine. A steam-gauge and thermometer were located on the pipe, through which all steam condensed in the jackets flowed to the Nason trap, at a point about 3 feet from the centre line of the engine.

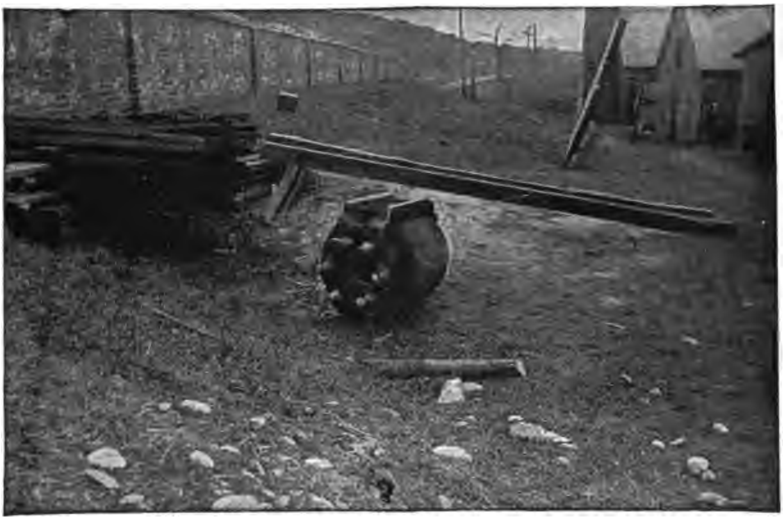


FIG. 287.

Showing yard containing boiler, engine and telegraph houses and machine shop; also, in the foreground, the jacket or sleeve clamped on the outside of the line pipe, over a break or rupture in the latter.

A rubber packing ring is squeezed together upon the pipe at each end of the casting, and a gland or stuffing-box attached by means of the stud bolts shown. This device enables a leak to be controlled without removing the ruptured length of pipe from the line.

DESCRIPTION AND CALIBRATION OF STROKE REGISTER.

One of these instruments is shown in Fig. 288. It consists of a ratchet-wheel attached to the primary arbor of the train of gear-wheels, and indices, constituting the recording mechanism of water meters. Mounted on the same arbor with the ratchet-wheel is a pinion gearing into a rack attached to the sliding piece A, and the latter carries a Stubbs steel rod upon which are mounted the stops B B, which are adjustable, so that the projecting points C C may be set any desired distance apart. The teeth of the ratchet are so finely cut that by the aid of two pawls, set so as to subdivide a tooth-space, a movement of the rack as small as one-thirty-second of an inch will move the primary index of the recording mechanism. If, therefore, the apparatus is mounted so that an arm clamped to the cross-head of the pumping engine can strike the projecting points C C, and the distance of the latter apart be any amount, not more than two inches, less than the maximum stroke of the engine, the recording mechanism will register numbers representing the total travel of the engine, in excess of that corresponding to a length of stroke equal to the distance apart of the projecting points C C, plus the thickness of the arm attached to the cross-head. Thus, in the present case, the stroke of the engine was about 37¾ inches. The stops, or projecting points C C, were therefore set so that their distance apart was 36 inches, plus the thickness of the arm fastened to the cross-head of the engine. The proportions of the ratchet and rack motion were such that a movement of the rack equal to five-ninths of an inch moved the primary hand of the recording mechanisms one unit of its dial, or one-tenth of a revolu-

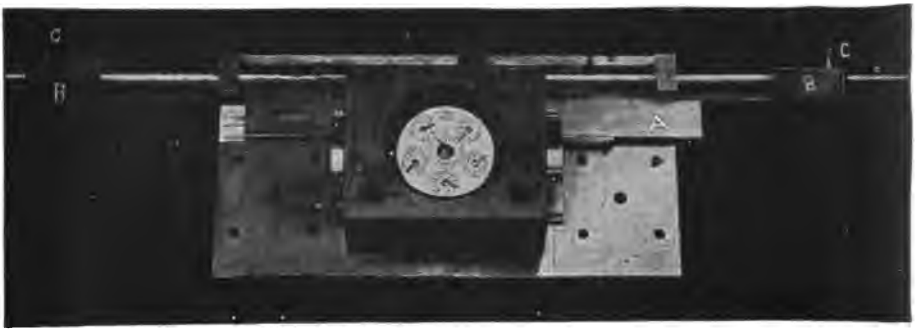


FIG. 288.

tion. Hence the average length of stroke made by the engine was given as follows:

Let
N_1 = Initial reading of revolution counter of engine.
N_2 = Final " " " "
R_1 = Initial reading of stroke register.
R_2 = Final " " "
Then average length of stroke in inches is

$$L = 36 + \frac{5\ (R_2 - R_1)}{9\ (N_2 - N_1)}$$

The instruments were calibrated by moving the rack a known number of times between stops a measured distance apart, before attachment to the engine. This calibration was also checked by several series of observations of the movement of the rack A, while attached to the engine. The rack being at rest while the piston of the engine is traveling 36 inches, a scale can be placed beneath it, and the excess of piston travel, over 36 inches, be easily read off each stroke. Examples of such a series of observations, one for each side of the engine, are given below.

LENGTH OF STROKE IN EXCESS OF 36 INCHES FOR 100 DIVISIONS OF PRIMARY INDEX.

North Side.				
No. of Double Strokes.	Rack Movement. Inches.	No. of Double Strokes.	Rack Movement. Inches.	
1	$1\frac{8}{16}$	19	$1\frac{9}{16}$	Total rack movement, 54.53 inches; or roundly, $\frac{5}{8}$ inch per division of primary index.
2	$1\frac{9}{16}$	20	$1\frac{8}{16}$	
3	$1\frac{9}{16}$	21	$1\frac{9}{16}$	
4	$1\frac{10}{16}$	22	$1\frac{10}{16}$	
5	$1\frac{10}{16}$	23	$1\frac{9}{16}$	
6	$1\frac{10}{16}$	24	$1\frac{9}{16}$	
7	$1\frac{9}{16}$	25	$1\frac{9}{16}$	
8	$1\frac{10}{16}$	26	$1\frac{9}{16}$	
9	$1\frac{1}{16}$	27	$1\frac{8}{16}$	
10	$1\frac{10}{16}$	28	$1\frac{7}{16}$	
11	$1\frac{8}{16}$	29	$1\frac{8}{16}$	
12	$1\frac{9}{16}$	30	$1\frac{8}{16}$	
13	$1\frac{10}{16}$	31	$1\frac{7}{16}$	
14	$1\frac{10}{16}$	32	$1\frac{8}{16}$	
15	$1\frac{9}{16}$	33	$1\frac{7}{16}$	
16	$1\frac{10}{16}$	34	$1\frac{8}{16}$	
17	$1\frac{1}{16}$	35	$1\frac{8}{16}$	
18	$1\frac{9}{16}$	36		

LENGTH OF STROKE IN EXCESS OF 36 INCHES FOR 150 DIVISIONS OF PRIMARY INDEX.

SOUTH SIDE.				
No. of Double Strokes.	Rack Movement. Inches.	No. of Double Strokes.	Rack Movement. Inches.	
1	$1\frac{5}{16}$	32	$1\frac{5}{16}$	Total rack movement, 81.3 inches; or roundly, $\frac{5}{9}$ inch per division of primary index.
2	$1\frac{4}{16}$	33	$1\frac{5}{16}$	
3	$1\frac{3}{16}$	34	$1\frac{4}{16}$	
4	$1\frac{4}{16}$	35	$1\frac{4}{16}$	
5	$1\frac{5}{16}$	36	$1\frac{6}{16}$	
6	$1\frac{4}{16}$	37	$1\frac{4}{16}$	
7	$1\frac{5}{16}$	38	$1\frac{5}{16}$	
8	$1\frac{5}{16}$	39	$1\frac{6}{16}$	
9	$1\frac{5}{16}$	40	$1\frac{6}{16}$	
10	$1\frac{5}{16}$	41	$1\frac{4}{16}$	
11	$1\frac{4}{16}$	42	$1\frac{5}{16}$	
12	$1\frac{5}{16}$	43	$1\frac{5}{16}$	
13	$1\frac{4}{16}$	44	$1\frac{5}{16}$	
14	$1\frac{5}{16}$	45	$1\frac{5}{16}$	
15	$1\frac{5}{16}$	46	$1\frac{6}{16}$	
16	$1\frac{6}{16}$	47	$1\frac{5}{16}$	
17	$1\frac{5}{16}$	48	$1\frac{4}{16}$	
18	$1\frac{6}{16}$	49	$1\frac{4}{16}$	
19	$1\frac{4}{16}$	50	$1\frac{4}{16}$	
20	$1\frac{5}{16}$	51	$1\frac{5}{16}$	
21	$1\frac{4}{16}$	52	$1\frac{6}{16}$	
22	$1\frac{4}{16}$	53	$1\frac{5}{16}$	
23	$1\frac{4}{16}$	54	$1\frac{5}{16}$	
24	$1\frac{4}{16}$	55	$1\frac{9}{16}$	
25	$1\frac{4}{16}$	56	$1\frac{6}{16}$	
26	$1\frac{3}{16}$	57	$1\frac{6}{16}$	
27	$1\frac{4}{16}$	58	$1\frac{9}{16}$	
28	$1\frac{4}{16}$	59	$1\frac{7}{16}$	
29	$1\frac{5}{16}$	60	$1\frac{6}{16}$	
30	$1\frac{7}{16}$	61	$1\frac{4}{16}$	
31	$1\frac{6}{16}$	62	$1\frac{6}{16}$	

DESCRIPTION AND CALIBRATION OF INDICATOR APPLIED TO PUMP CYLINDERS.

This consisted of a steel plunger one-quarter of an inch in diameter, which received the pressure of the pump cylinders against its lower end through the medium of a mass of cylinder oil, and communicated this pressure to the piston of a Crosby indicator, against which the upper end of the plunger abutted. The apparatus, seen as adjusted for calibration and on the engine, respectively, is shown in Figs. 289 and 284. The brass syphon A contained the cylinder oil, and the quarter-inch plunger works oil-tight through the hexagonal nut B. A three-way cock at C is set so that there is communication through it, with the atmosphere, while communication toward the pump cylinders is closed. The cock D being then opened, a charge of cylinder oil was introduced with a syringe, until this oil was seen escaping to the atmosphere at C.

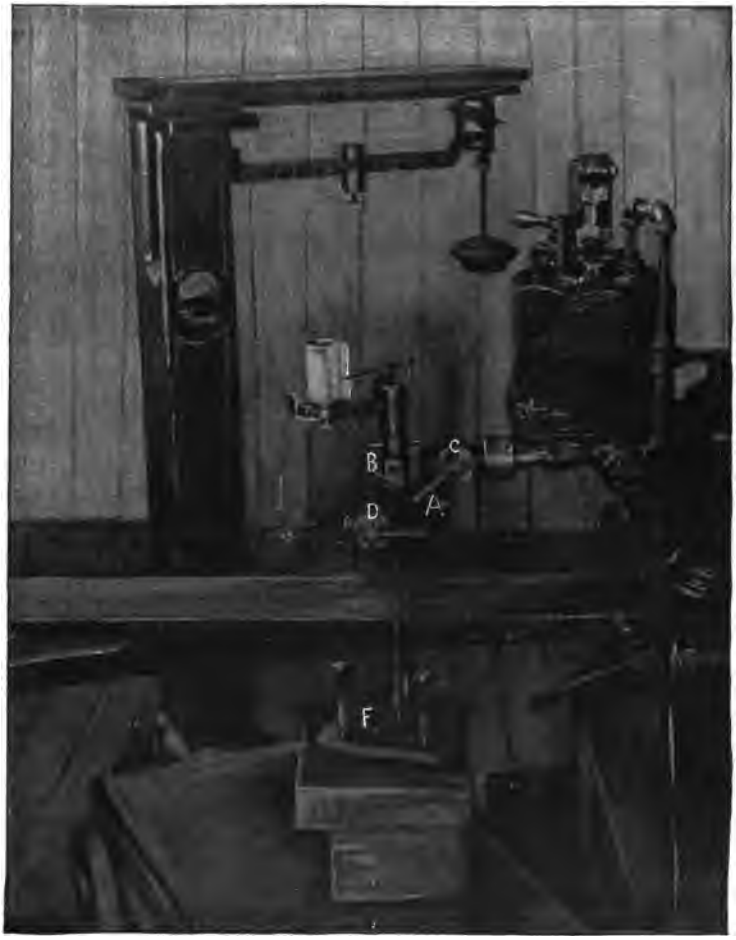


FIG. 289.

INDICATOR FOR PUMPS.

The cock D is then closed, and C turned so as to communicate with the cylinders, and allow the pump pressure to act upon the plunger.

There was practically no leakage with the plunger fitted a thousandth of an inch less in diameter than the hole in the nut B. The instrument was filled, or recharged, only when it was thought that the crude oil from the pump cylinders may have taken the place of the cylinder oil, through the escape of the latter by the frequent setting of the cock C to connect with the atmosphere, in taking the atmospheric lines of cards. To calibrate the instrument, it was emptied of oil and a plug directly under the plunger removed. The brass rod E was then adjusted so that, by working the leveling screws in the platform F, the indicator pencil was moved over its range of travel. The average readings of the platform scale, corresponding to each sixth of an inch of pencil travel, gave the scale of the instrument as 610 pounds per inch.

As a check upon this method of calibration, the spring in the Crosby instrument and a small spring attached to the plunger, to enable it to measure pressures below the atmosphere, were calibrated separately, and gave the same value for the scale of the instrument.

It is probable that this value is correct to within about 1½ per cent. As no mercury column exists which affords a means of calibrating pressure gauges to pressures as high as 900 pounds per square inch, this indicator was the sole basis of oil pressure measurement available, and the several Bourdon gauges used for recording the oil pressure were all referred to this instrument. Both the stroke register and the hydraulic indicator were designed, at the request of the writer, by Prof. J. B. Webb. The steam indicators were calibrated from 27 inches of vacuum to 20 pounds above the atmosphere, with mercury columns, and for higher pressures, by means of the "dead weight test-plate" of the Utica Gauge Company. In the course of the experiments seven Ashcroft indicators and one Thompson indicator were used. Four of the Ashcroft were new, being kindly loaned the writer by the Ashcroft Company. The scales of the latter were found correct to within 1 per cent. The water meters were all calibrated in place, under the exact conditions of pressure and rate at which they were used. Calibrations were made before and after the experiments, and the variations amounted to only four-tenths of 1 per cent.

PRINCIPAL DIMENSIONS OF ENGINE AND BOILERS.

The boilers were of the return tubular type, the gases passing once under shell, once through the tubes, and thence to the chimney.

Diameter of shell,	5 ft.
Number of tubes, each boiler,	82
Diameter of tubes,	3 in.
Length of boiler and tubes,	14 ft.
Width of grate,	4 ft. 8 in.
Length of grate,	6 ft.
Grate surface per boiler,	28 sq. ft.
Heating surface in tubes per boiler,	895 "
Heating surface in shell, "	111 "
Total heating surface "	906 "
Superheating surface,	none.
Ratio of heating to grate surface,	32

The engine was of the tandem compound type, with a receiver between each pair of high and low cylinders, the steam being heated while passing through a receiver by contact with a nest of piping filled with steam at boiler pressure. (See Fig. 291.) Both the barrels and heads of all cylinders were jacketed with steam taken directly from the boiler. Steam for the jackets and the reheater in receiver was taken from the main steam-pipe by an inch and a-quarter pipe, leaving the bottom of the latter immediately under the main throttle valve. (See Fig. 285.) From this pipe, branch pipes, one and one-quarter inches diameter, led steam to the reheater and jackets. The drainage from the reheater passed through the cylinder jackets, and the drains from the latter all united in a single pipe underneath the cylinders, leading to a Nason trap in the case of the test, but to an automatic steam return trap on top of the boilers, in the ordinary running of the station. The pump plungers were of the outside-packed style, working through stuffing boxes packed with rubber and hemp. To distinguish between the two sides of the engine, we designate as the No. 1 engine the left-hand side looking from the steam toward the pump end; and as No. 2 engine, the right-hand side from the same stand-point.

DIMENSIONS OF ENGINES.

Diameter of both high-pressure cylinders,	33 inches
" low-pressure cylinders,	66 "
" four-pump cylinders,	9¼ "
" low-pressure piston rod,	5¼ "
" high-pressure piston rod,	5½ "
" piston of both air pumps, double acting,	10½ "
Average stroke of air pumps,	37½ "

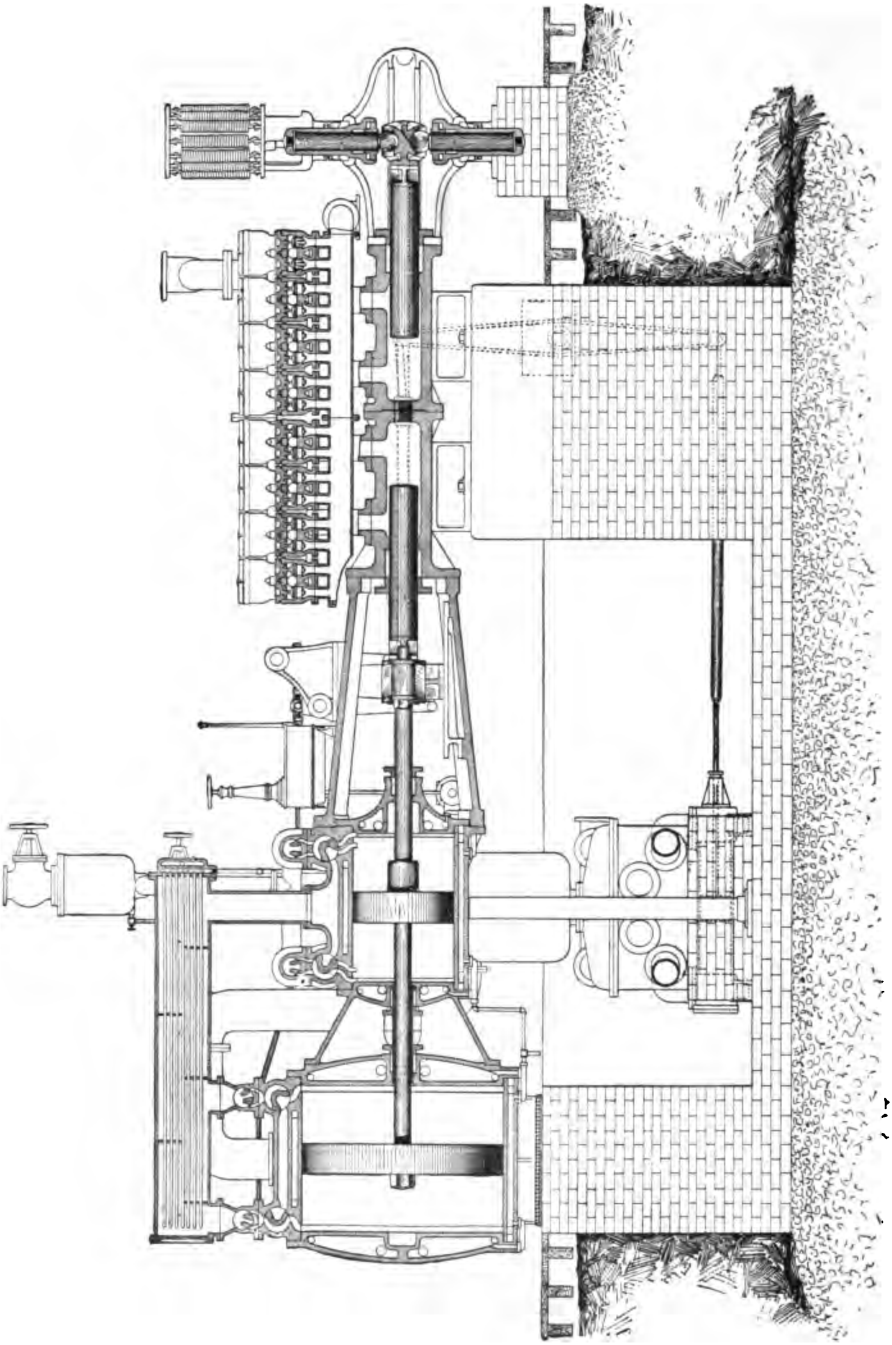


FIG. 291

Total possible travel of No. 1 engine pistons,	$37\frac{84}{100}$ in.
Distance between prick punch marks, No. 1 engine,	$37\frac{5}{8}$ “
Average travel determined by stroke register,	$37\frac{34}{100}$ “
Total possible travel, No. 2 engine pistons,	$37\frac{85}{100}$ “
Distance between prick punch marks, No. 2 engine,	$37\frac{79}{100}$ “
Average travel determined by stroke register,	$37\frac{71}{100}$ “
Ratio of high piston area to cut-off port area,	.021
Ratio of high piston area to exhaust port area,	.0468
Ratio of low piston area to cut-off port area,	.0145
Ratio of low piston area to exhaust port area,	.0211
Net area, each low piston,	3,410 sq. in.
Net area, each high piston,	832 “
Area of each pump plunger,	$67\frac{2}{10}$ “
Ratio of low piston area to high piston area,	4.097
Ratio of low piston area to pump plunger area,	$50\frac{7}{10}$
Clearance of high piston in per cent. of average piston displacement, both engines:	
Cut-off ports,	0.52 per ct.
Outside of exhaust valve,	0.79 “
Exhaust ports,	0.75 “
At ends of cylinders—average both engines,	0.45 “
Total average clearance, high cylinders,	2.51 per ct.
Clearance of low piston in per cent. of average piston displacement, both engines:	
Cut-off ports,	0.31 per ct.
Outside of exhaust valve,	$\frac{5}{8}$ “
Exhaust ports,	0.44 “
At ends of cylinders—average both engines,	0.45 “
Total average clearance, low cylinders,	$1\frac{7}{8}$ per ct.
Live steam heating surface in receiver, each engine,	228 sq. ft.
Ratio of cubic contents of receiver space, each engine, (to contents of high cylinder),	0.7
Number of inlet pump valves, each engine,	8
Total area inlet pump valves, each engine,	40 sq. in.
Area of inlet valves in per cent. of plunger area,	59 per ct.
Number of outlet valves, each engine,	8

Total area of outlet valves, each engine, - - -	40 sq. in.
Diameter of compensating piston, - - - -	6 in.
Stroke of compensating piston, - - - - - -	9 "
Diameter of trunnions of compensating cylinders, -	6 "
Diameter of thrust bearing of compensating piston, -	7 "
Diameter of main steam supply pipe, - - - -	6 "
Length of main steam supply pipe, - - - -	100 ft.
Cubic contents of foundation, both engines, - -	980 cu. ft.

Results of Experiments.

PERFORMANCE OF BOILERS.

Four boiler tests were made, as follows:

First.—A 24-hour test, April 1st, using pea coal, and six boilers all connected, no separation being made of the steam used by the various motors.

Second.—A 24-hour test, April 3d, using pea coal, five boilers being devoted to the pumping engine and a sixth to the other motors.

Third.—A 24-hour test, April 6th, using anthracite stove coal, five boilers being used for the engine, and one for the other motors.

Fourth.—A 12-hour test, April 8th, using anthracite stove coal, four boilers being used for the engine, and one for the other motors.

The fires were not drawn in starting the boiler test, but the time of starting was made to coincide with the hour when the fires were all cleaned, 1 o'clock—the regular habits of the firemen being regarded as insuring as much accuracy, by this plan, as would ensue from attempting to draw the fires of so many boilers, in order to start with wood.

The single boiler had to be operated with the fire doors open. It evaporated about 410 pounds of water per hour, with an economy of 6¼ pounds of water per pound of coal at actual steam pressure and from actual feed-temperature. The steam consumed by the feed-pump was probably about 1 per cent. of that used by the main engine.

BOILER ECONOMY.

	Pea Coal.		Anthracite Coal	
	First Test.	Second Test.	First Test.	Second Test.
Date of test	April 1.	April 3.	April 6.	April 8.
Duration of test	24 hours	24 hours	24 hours	24 hours
Boilers in use under equal conditions	6	5	5	4
Pounds consumed per hour	940	865	839	800
Pounds water evaporated per hour from actual temperature of feed and at actual boiler pressure	8225	7574	7489	7523
Average temperature of feed, degrees Fahr	118	119	113	116
Average boiler pressure	89	90	90	89
Pounds actual evaporation per pound of coal	8.78	8.74	8.91	9.46
Ashes, per cent	......	....	10.0	11.0
Temperature of feed-water if jacket water returned to boiler as in usual working of station, degrees Fahr	147	151	146	147
Pounds evaporation per pound of coal for 147° temperature of feed and actual steam pressure	9.09	9.04	9.22	9.79
Pounds evaporation per pound of coal from and at 212° Fahr	9.94	9.89	10.14	10.71
Pounds evaporation per pound of combustible from and at 212°	....	......	11.26	11.91
Chimney temperature, degrees Fahr	340	360	390	410
Condition of steam	Dry	Dry	Dry	Dry
Pounds coal consumed per square foot of grate per hour	5.6	6.2	6.0	7.1
Pounds water evaporated per hour per square foot of heating surface from and at 212° Fahr	1.77	1.89	1.88	2.25

This corresponds to a duty of 10,000,000 foot-pounds, and an allowance of 5 per cent. for friction. If the exhaust of the feed-pump was absorbed in the feed-water, the temperature of the latter would be increased about 10 degrees. The expense of operating the feed-pump would then increase the steam-consumption of the main engine one-eighth of 1 per cent.

The boiler-flues were cleaned on March 28th and on April 4th. The interior surfaces of the boilers were clean, and quite free from scale.

There was no leakage at the blow-off valves of the boilers, but some of the safety-valves leaked slightly.

PERFORMANCE OF ENGINE.

No attempt was made to adjust the engine especially for testing. It was run under exactly the conditions of its operation for the previous six months, without any extra attendance or care. No test was made to determine the leakage of the valves or pistons.

The engine differed from the standard construction of Worthington Water Works Engine in the following details instituted by the pipe-line management.

First.—The admission-valves of both the high and low cylinders were fitted with strips lying at an angle with the edge of the port so as to cause the latter to open gradually.

Second—The steam-cylinders were lagged simply with a 2-inch layer of asbestos covered with tarred canvas.

ENGINE ECONOMY.

THROTTLE VALVE AND VALVE BETWEEN RECEIVERS WIDE OPEN.

1	Date of test, - - - - - - - - -	April 3.	April 6.
2	Duration of test, - - - - - - -	24 hours.	24 hours.
3	Average steam pressure at boiler, in pounds per square inch above atmosphere, - -	89	89
4	Average steam pressure at engine, in pounds, per square inch above atmosphere, - -	88	89
5	Average steam pressure under main throttle, in pounds, per square inch above atmosphere, - - - - - - . - -	84	85
6	Average steam pressure under branch throttle, in pounds, per square inch above atmosphere, - - - - - - - - -	82	83
7	Average steam pressure at bottom of steam jackets, in pounds, per square inch above atmosphere, - - - - - - - -	82	83
8	Average steam pressure during admission to high cylinder, in pounds, per square inch above atmosphere, - - - - - -	74	74
9	Average oil pressure at delivery, in pounds, per square inch above atmosphere, - -	872	858
10	Average oil pressure at suction, in pounds, per square inch above atmosphere, - - -	0	0
11	Average vacuum, inches of mercury, - -	27.25	27.25
12	" barometric pressure, inches of mercury, - - - - - - - - -	29.75	29.75
13	Average temperature outlet to jackets, degrees Fahr., - - - - - - - - - -	325	326
14	Average temperature feed-water, - - -	119	113
15	Temperature of feed-water, if drainage from jackets had returned to the boilers, degrees Fahr., - - - - - - - - -	151	146
16	Thermal units per pound of steam at boiler pressure, - - - - - - - -	1,214	1,214

17 Thermal units to evaporate 1 pound of steam, if jacket water was returned to boiler, -	1,064	1,068
18 Total pounds steam supplied engine per hour,	7,574	7,489
19 Per cent. of latter condensed by jackets and reheater, - - - - - - - - -	15.5	16.1
20 Average length of stroke in feet, - - -	3.127	3.128
21 " revolutions per minute, - - -	20.16	20.13
22 " feet of piston travel per minute, -	126	125.9
23 " horse-power at pump end, - - -	446.4	439.9
24 Pounds of steam condensed per hour per pump horse-power, - - - - - - -	16.96	17.02
25 Pounds of coal per hour per pump horse-power, assuming 10 pounds of steam to be evaporated per pound of coal, - - -	1.696	1.702
26 Foot-pounds duty per 100 pounds of coal at 10 pounds evaporation, - - - - -	116,730,000	116,330,000
27 Foot-pounds duty per 1,000,000 thermal units of steam consumption, neglecting steam consumed by feed pump, - - - -	110,000,000	109,000,000
28 Horse-power to operate feed pump, - - -	0.9	0.9
29 Foot-pounds duty per million heat units of steam consumption, including steam consumed by feed-pump, assuming latter to give a duty of 10,000,000 foot-pounds under actual conditions, and to exhaust into the feed-water, - - - - - -	109,850,000	108,860,000
	Pea coal.	Anthracite coal.
30 Foot-pounds duty per 100 pounds of coal at average rate of evaporation, under working conditions, - - - - - - - -	105,395,000	110,943,000
31 Thermal units of steam consumption per pump horse-power per minute, - - - - -	300	302
32 Average mean effective pressure, high cylinders, pounds per square inch, - - - - - - - - -		34.08
33 Average mean effective pressure, low cylinders, pounds per square inch, - - - - - - - - -		9.02
34 Average mean effective pressure, all cylinders, per square inch of low piston, - - - - - - - - -		17.53
35 Total horse-power, steam end, - - - - - -		456.7
36 Pounds of water consumption per steam horse-power by measurement, - - - - - - - - - -		16.40
37 Friction of mechanism per cent. of steam horse-power, -		3.6
38 Thermal units of steam consumption per steam horse-power per minute, - - - - - - - - -		291
39 Average apparent cut-off high cylinder, - - - -		0.33
40 Total real expansions by volume, both cylinders, - -		13.76
41 Per cent. of total feed-water not accounted for at cut-off high cylinder, - - - - - - - - - -		34.0
42 Ditto at release of high cylinder, - - - - -		30.0

43	Ditto at cut-off of low cylinder,	12.0
44	Ditto at release of low cylinder,	12.0
45	Moisture in steam entering engine,	1.1%
46	Oil consumed in cylinders, gallons per 24 hours,	1.0
47	Oil consumed on outside bearings, gallons per 24 hours,	1.5

NOTES REGARDING FIGURES IN ABOVE TABLE.

First.—The oil pressure given in item 9 represents both the mean effective pressure in the pump cylinders, and the pressure shown by a gauge attached to the delivery chamber of the pump; because there was no practical pressure required to operate the delivery valves. This fact is shown by the sample pump cards (Fig. 292), in which the full lines were taken with the indicator attached to the pump cylinders, or inside the delivery valves, and the dotted line, with the indicator attached to the same pipe as the station pressure gauge, which shows the pressure in the delivery chamber of the pump, or outside the delivery valves.

CARDS TAKEN AT OIL END OF ENGINE.

Scale, 610 lbs. = 1 inch.

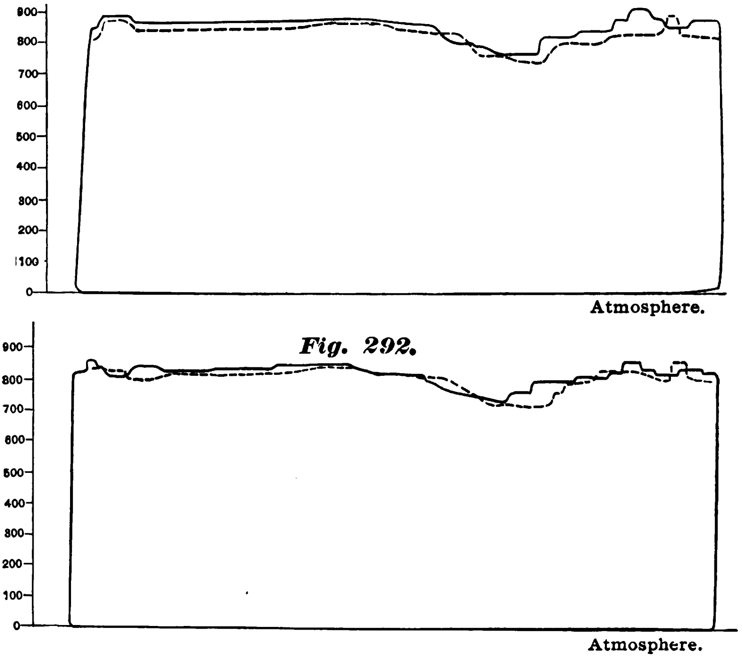


Fig. 292.

Full line—Indicator connected to pump cylinder inside of outlet valves.
Dotted line—Indicator connected to delivery chamber outside of outlet valves.

Second.—The duty, item 25, is calculated as follows: Test of April 6th.

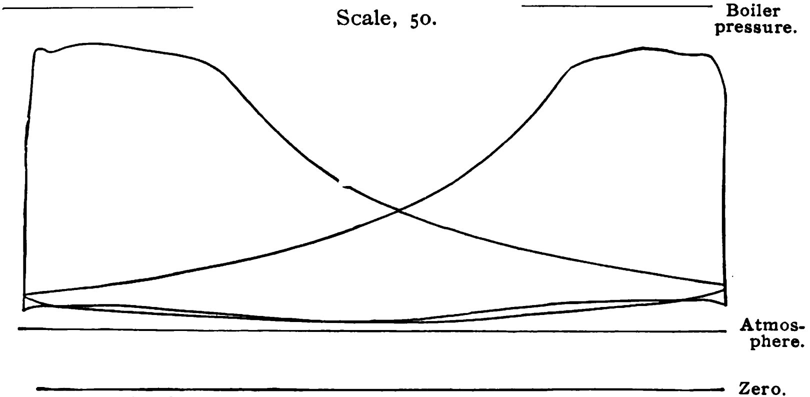
Oil press.		Area plungers.		No. plungers.		Revs. per min.		Stroke.		Pump H. P.
858	×	67.2	×	4	×	20.13	×	3.127	=	439.9
				33,000						

Water per hour.		Pump H. P.		Water per hr. per pump H. P.
7.489	÷	439.9	=	17.02 lbs.
Water per pump H. P.		Assumed evaporation per lb of coal.		Coal per hr. per pump H. P.
17.02	÷	10	=	1.702 lbs.
Ft. lbs. per hr. for one H. P.		Coal per hr. per pump H. P.		Duty per one lb. of coal.
1,980,000	÷	1.702	=	1,163,300
				Duty per 100 lbs. of coal.
1,163,300	×	100	=	116,330,000
		Ratio, item 17, to 1,000,000		Duty per 1,000,000 thermal units.
116,330,000	÷	1.068	=	109,000,000

Third.—The steam indicator cards were taken from both ends of each cylinder with one indicator, located at a tee midway between the ends.

TEST OF APRIL 6TH.

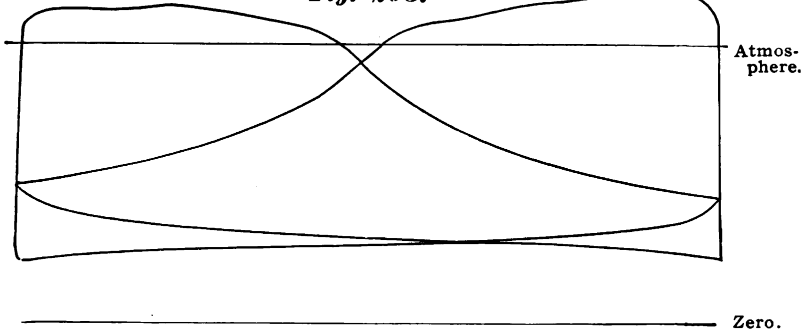
Scale, 50.

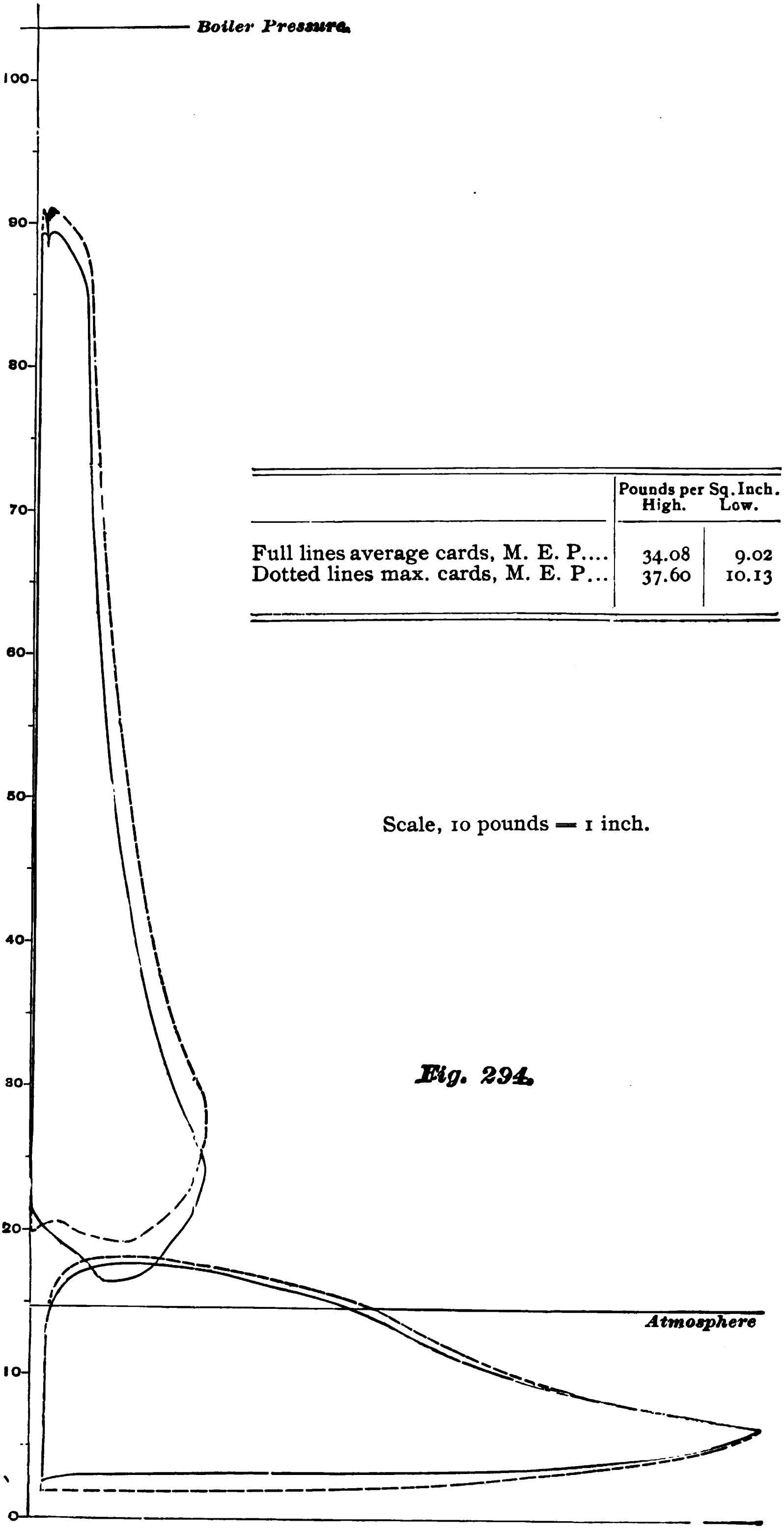


Cards of average M. E. P. combined in full lines, Fig. 294.

Scale, 10.

Fig. 293.





	Pounds per Sq. Inch.	
	High.	Low.
Full lines average cards, M. E. P....	34.08	9.02
Dotted lines max. cards, M. E. P...	37.60	10.13

Scale, 10 pounds = 1 inch.

Fig. 294.

TEST OF APRIL 6TH.

Scale, 50.

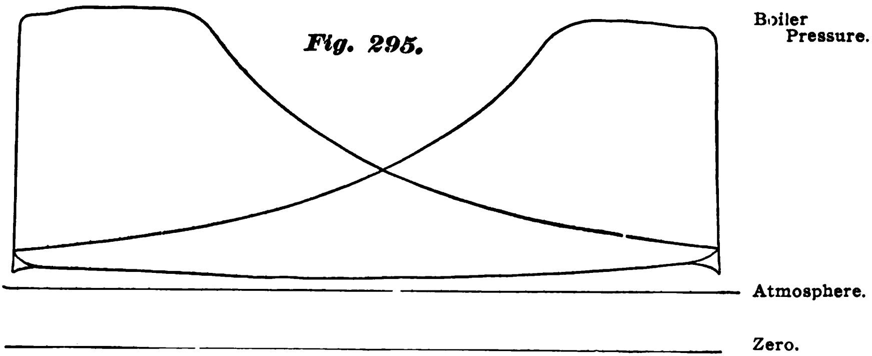
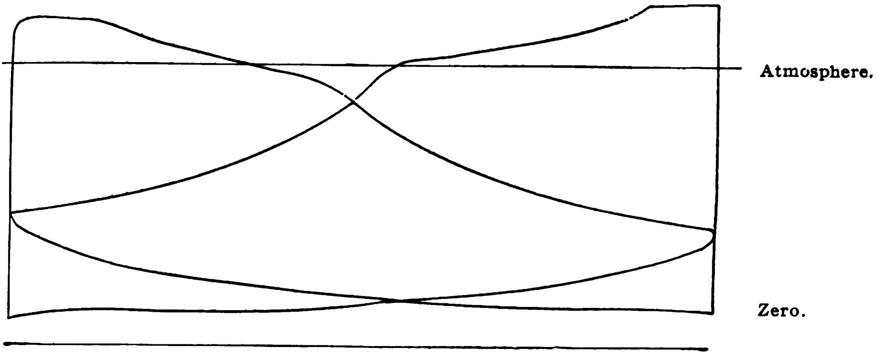


Fig. 295.

Cards of best form combined in Fig. 296.

Scale, 10.



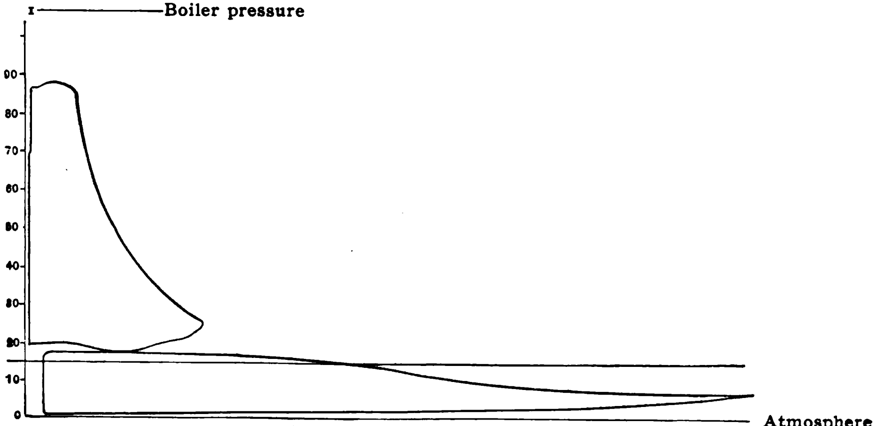


Fig. 296.

Scale, 50 pounds = 1 inch.

Such an arrangement, with high speeds of rotation, is inadmissible, but it was determined in the present instance that the arrangement gave exactly the same results as the use of separate indicators at each end of the steam-cylinders, with only a straight half-inch nipple between the indicator cock and the interior of the cylinder.

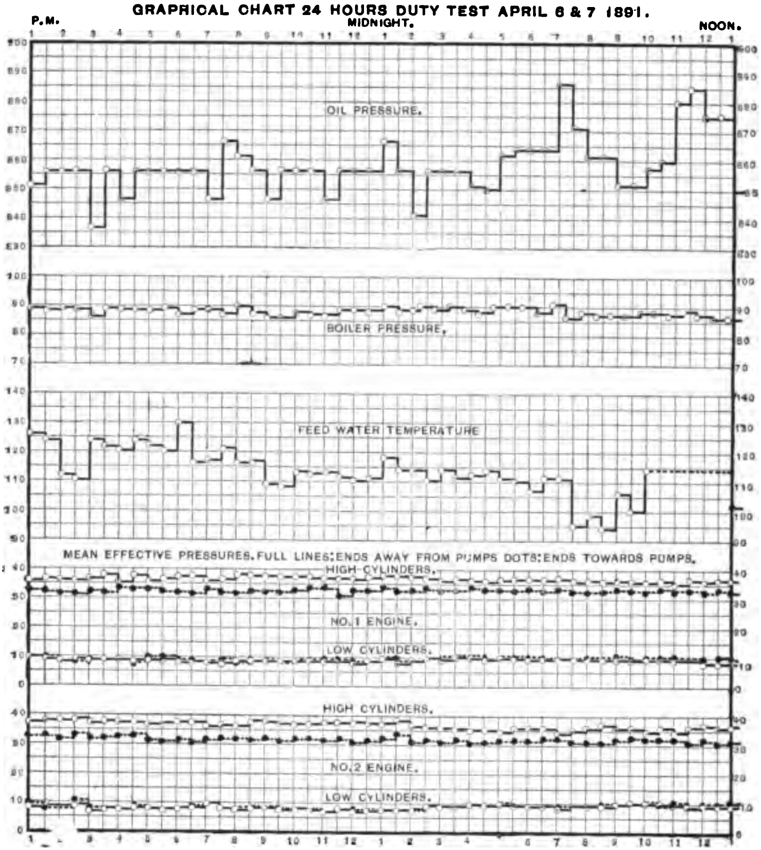


Fig. 297.

Sample indicator cards are shown in Figs. 293, 294, 295, 296; and their range of variation in mean effective pressure is shown by the graphical log, Figs. 297 and 298, and the tables in Appendix. It may be noticed that the variation in mean effective pressure is comparatively large, amounting to about 10 per cent. for the high cylinder and 15 per cent. for the low cylinder. Such variations are believed to accurately represent the accidental discrepancies to which the best work with indicators is subject. The effect of the irregularities is not great, however, with so many observations. The method of least squares shows that the probable error of the arithmetical mean of the mean effective pressures, reduced to the

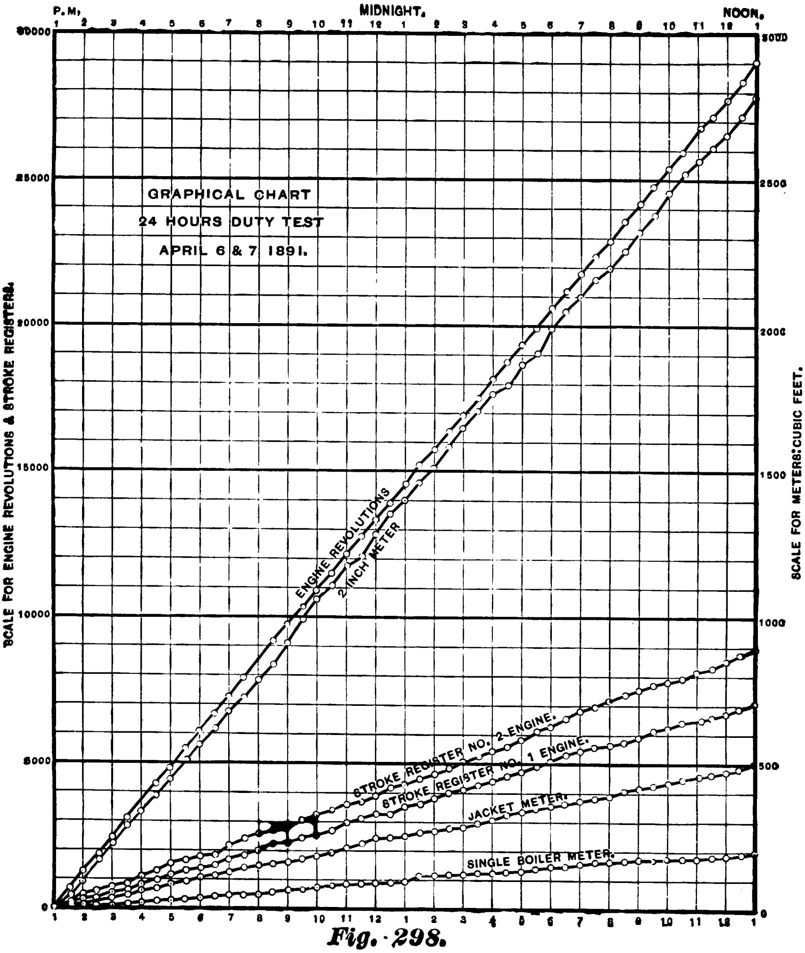


Fig. 298.

low piston, only aggregates 1 per cent., and none of the values are rejectable as abnormally large or small, according to *Pierce's Criterion.* The mean effective oil pressure is subjected to the same amount of error, due to variations of value, and the scale of the oil indicator is believed liable to an error of 1½ per cent. Such errors may be either plus or minus, and may partly neutralize or magnify each other's effects in the main result. In the present instance, as the average steam-cards give a very low value of friction, it has been judged proper to consider the true steam horse-power as that due to the arithmetical mean of the steam mean effective pressures, increased by 1 per cent. Also, since the value used for the scale of the hydraulic indicator, 610 pounds per inch, is probably on the low side of the truth, the true pump power is taken as that due to exactly the arithmetical mean of the mean effective pressures of the pump cylinders.

Fourth.—The friction of the mechanism, item 33, is considerably less than has been found in measurements of inside plunger Worthington pumps, and, hence, the pumps, in this instance having two outside packed stuffing-boxes on each side, might naturally be expected to waste more power in friction than an outside plunger working freely in the midst of water. But the plunger and oscillators, in the case of pumping oil, lubricate themselves so completely that, when the packing is so nicely and permanently adjusted as in the present case, the friction of the several plungers may be very little. A computation of the power required to operate the air pumps, assuming their mean effective pressure to be 6½ pounds, as usually shown by air-pump cards, gives three-fourths of 1 per cent. of the steam horse-power.

A computation of the friction of the several moving parts of the engine, based upon allowances which have been found to give correct results in similar estimates,* indicates that their aggregate friction might easily be only 3 per cent. of the steam power. Finally, by equalizing the pressure on both sides of the pumps, by means of a by-pass, and taking the mean pressure which, exerted on the high piston only, will just move the pump, such pressure corresponds to less than 5 per cent. of the steam power. The friction of the pump plungers under these circumstances is greater than while running, as the lubrication is practically destroyed. It is, therefore, concluded† that the 3.6 per cent. of friction found is an entirely reasonable amount.

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* See "Computation of Friction," of Pawtucket pumping engine and other motors, *Stevens Indicator*, Vol. VII., Nos. 1 and 2.

† A study of the possibility of the variations of pressure of the oil, during a stroke, causing the oscillating cylinders to act as a slight back pressure on the pumps, results in the conclusion that such action cannot exert such an effect of more than three-fourths of 1 per cent., if any whatever.

Fifth.—The moisture in the steam, item 41, is the average of several determinations made at different times when the calorimeter was observed for an interval of about an hour and a-half. The variation of the different tests was only a fraction of 1 per cent.

The moisture given, 1.1 per cent., is fairly attributable to the radiation, etc., of the steam supply pipes, which, while well-covered, aggregated 100 feet of 6-inch, and 125 feet of 4-inch pipe.

DETERMINATION OF SLIP OF PUMPS.

On April 23d the pump drew oil steadily for six and one-half hours from one of the receiving tanks, which, isolated for several days, showed no leakage or gain of contents.

The contents of the tank for each foot of depth is known within one-twentieth of 1 per cent., and, combined with a record of the stroke register and revolutions of the engine, afforded an unusual opportunity to determine the "slip" of the pump. The results obtained are as follows:

Revolutions per minute for 6½ hours, - - - -	22.05
Average length of stroke, No. 1 engine, - -	37.42 in.
" " " " " 2 " - - - -	37.91 "
" " " both strokes, - - - - -	3.14 ft.
Aggregate area of four plungers, - - - - -	1.867 sq. ft.
Cubic feet displaced by plungers, per hour, - -	7758.07
Depth of tank "pump out" in 6½ hours, - -	7.896 ft.
Cubic contents of each foot of tank in barrels of 42 gallons each, 231 cu. in. per gallon, - -	1121.9
Cubic feet "pumped out" per hour, - - -	7652.44
Slip in cubic feet per hour, 7758.07 — 7652.44 = -	105.63
" " per cent. of plunger displacement, - -	1.36

As the leakage from the plungers was comparatively infinitesimal, this slip is due wholly to valve action.

BEHAVIOR OF ENGINE WITHOUT STEAM IN JACKETS AND REHEATER.

On April 4th steam was shut off from the jackets and reheater, and the latter emptied into the hot-well. The engine was then run one hour and a-half. The stroke was very irregular. If adjusted to run 2 inches short of striking the heads, a piston would often strike for a few strokes and then the stroke again shorten.

Cards taken fifteen minutes after the steam was shut off from jackets.

Full lines are cards taken five minutes before shutting off steam from jackets. Dotted lines are cards taken fifteen minutes after shutting off steam from jackets.

Pump under excellent control and making full stroke.

High-pressure cylinder, No. 2 engine.
Scale of spring: 49.5
Mean effective pressures:
Right........ 29 10
Left.......... 34.02

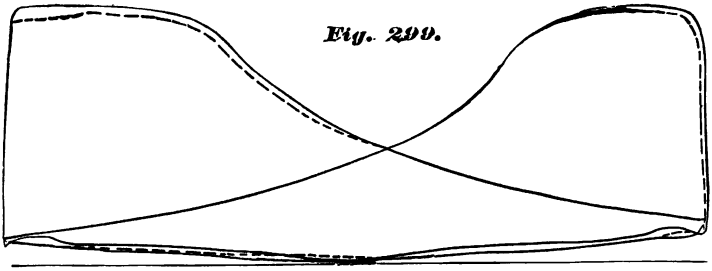


Fig. 299.

Low-pressure cylinder, No. 2 engine.
Scale of spring: 10.1
Mean effective pressures:
Right.... .... 8.38
Left....... . 8.18

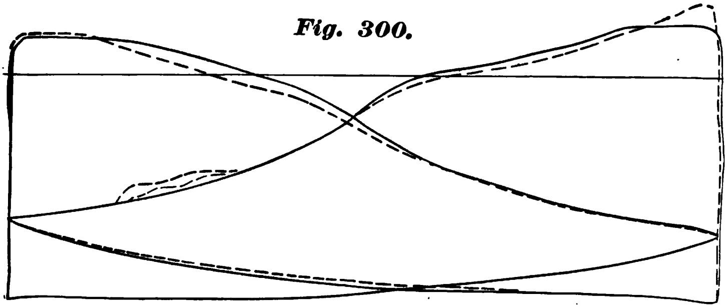


Fig. 300.

High-pressure cylinder, No. 1 engine.
Scale of spring: 49.5.
Mean effective pressures:
Right 28.6
Left......... ... 33.9

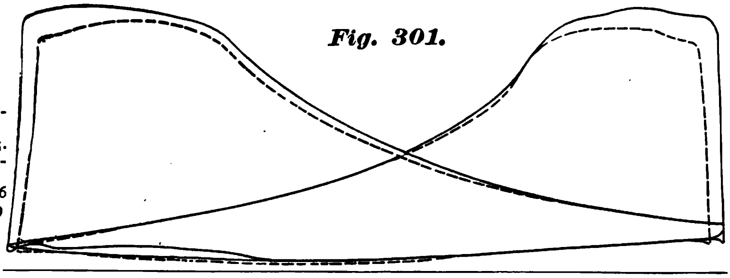
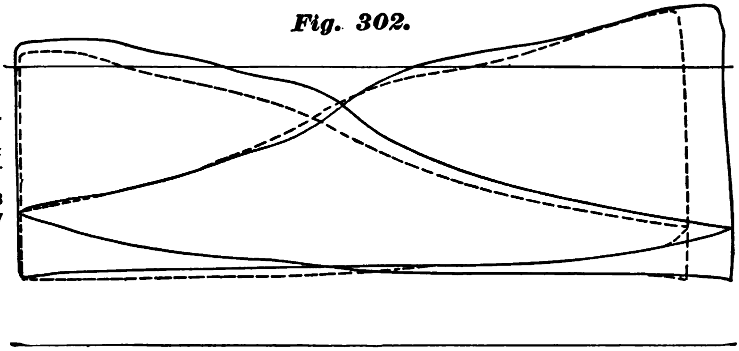


Fig. 301.

Fig. 302.



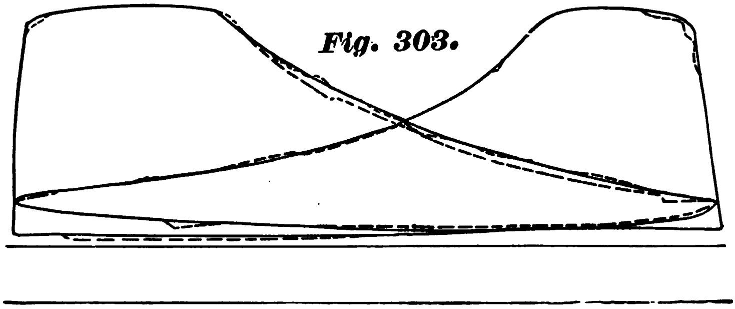
Low-pressure cylinder, No. 1 engine.
Scale of spring: 10.1
Mean effective pressures:
Right......... 8.28
Left......... 7.07

Cards taken twenty-five minutes after shutting off steam from jackets.

Dotted lines represent variations in the card while the pencil was in contact with the Paper one minute.

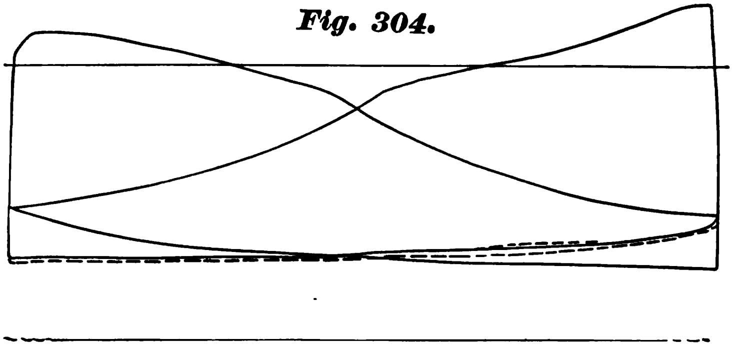
Pump under excellent control and making full stroke.

Fig. 303.



High-pressure cylinder, No. 2 engine.
Scale of spring: 49.5
Mean effective pressures:
Right........ 27.6
Left........... 33.9

Fig. 304.



Low-pressure cylinder, No. 2 engine.
Scale of spring: 10.1
Mean effective pressures:
Right.......... 7.27
Left........... 6.87

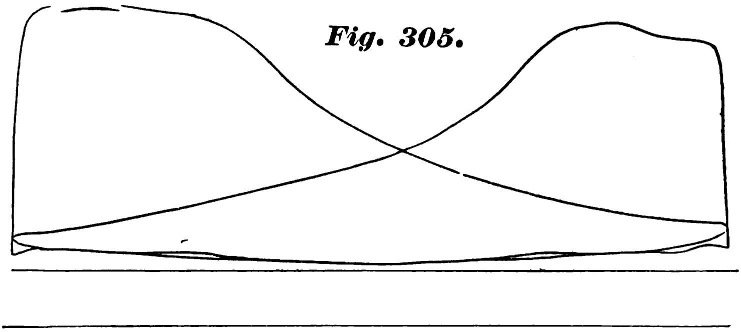


Fig. 305.

High-pressure cylinder, No. 1 engine.
Scale of spring: 49.5
Mean effective pressures:
Right.......... 29.1
Left........... 35.0

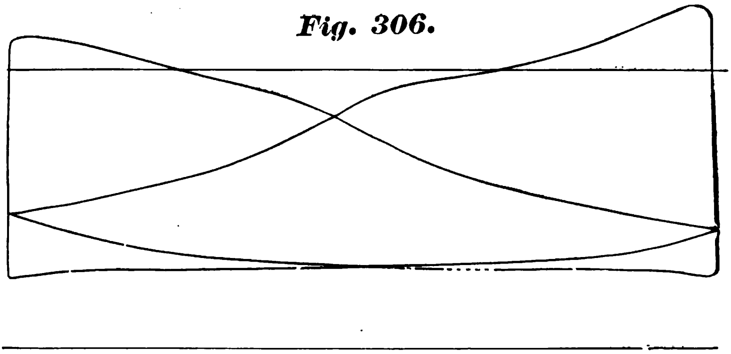


Fig. 306.

Low-pressure cylinder, No. 1 engine
Scale of spring: 10.1
Mean effective pressures:
Right.......... 7.57
Left 6.77

Cards taken forty-five minutes after shutting off steam from jackets.

Dotted lines represent the variations in the card while the pencil was in contact with the paper one minute.

Pump under excellent control and making full stroke.

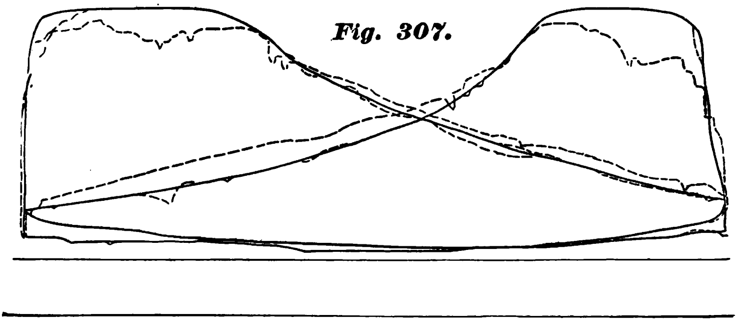
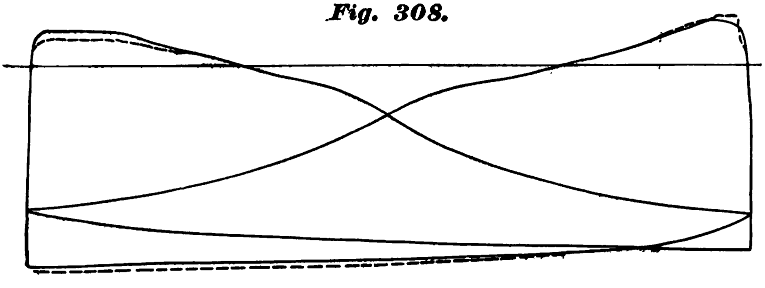


Fig. 307.

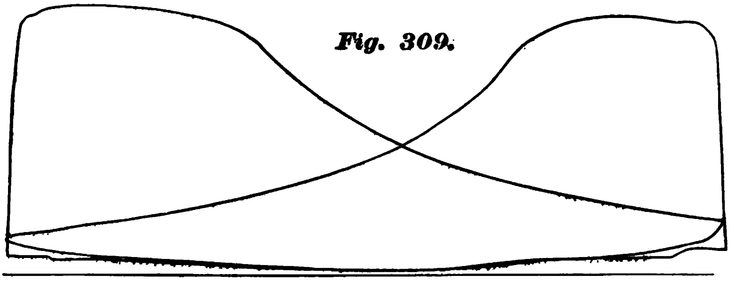
High-pressure cylinder, No. 2 engine.
Scale of spring: 49.5
Mean effective pressures:
Right............29.6
Left.............36.5

Fig. 308.



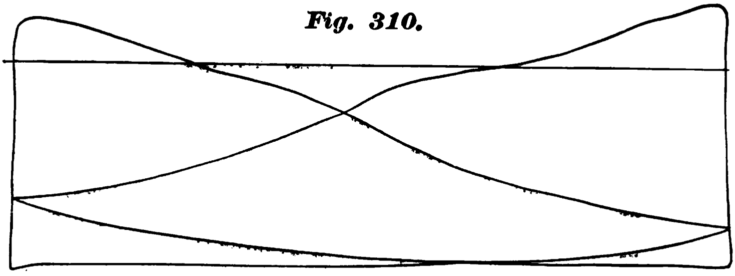
Low-pressure cylinder, No. 2 engine.
Scale of spring: 10.1
Mean effective pressures:
Right.......... 6.21
Left............7.17

Fig. 309.



High-pressure cylinder, No. 1 engine.
Scale of spring: 49.5
Mean effective pressures:
Right...........31.8
Left............37.5

Fig. 310.



Low-pressure cylinder, No. 1 engine.
Scale of spring: 10.1
Mean effective pressures:
Right...........7.37
Left............7.07

The revolutions dropped from 20.13 to 16.6 per minute, and the oil pressure from 850 to 750 pounds. The indicator cards were very irregular from water in the cylinders, but occasionally a well-formed card could be obtained. The water consumption per horse-power was increased 20 per cent., but, as less work was being done, the coal consumption did not noticeably increase.

On April 23d steam was again shut off the jackets and reheated, and, from the experience of the previous experiment, the engine was more quickly and nicely adjusted to the new conditions.

The stroke was easily maintained fairly regular for an hour, and the indicator cards, Figs. 299 to 310, obtained. The speed fell from 22.05 to 20.2 per minute, and the oil pressure a corresponding amount. No means of measuring the water consumption was then available.

The coal consumption did not noticeably change, but it is too coarse a means of measurement to be of any value in the case. After seven minutes of running, the water in one low cylinder could be heard to splash as the piston reached the end of its stroke. The vacuum fell from 27¼ to 23 inches. After running an hour, it was evident that the water in the cylinders was still increasing, although the pump was still under good control. Five minutes after restoring the steam to the jackets, the full vacuum had returned, the sound of the water in the cylinders disappeared, and the engine regained her speed. The indicator cards show about 20 per cent. less mean effective pressure in the low cylinders, and about equal mean effective pressure in the high cylinders compared with the cards taken with steam in the jackets and reheater. It was evident that, letting alone the question of economy of steam, steam-jackets are essential to the successful working of this style of engine.

CONCLUSIONS.

First.—The boiler economy of the station is equivalent to 9.1 and 9.8 pounds evaporation per pound of pea and stove anthracite, respectively, from the actual temperature of feed, 147° Fahr., and average steam pressure of 90 pounds above the atmosphere. It is probable that an evaporation greater than 10 pounds per pound of coal, under working conditions, could easily be obtained with the best bituminous coal.

Second.—The average economy of the engine is represented by any of the following statements:

(*a*) The steam consumed per hour per horse-power pump end is, - - - - -	17.0 lbs.
(*b*) The steam consumed per hour per horse-power steam end is, - - - - -	16.4 "
(*c*) The duty per 100 pounds of coal, if the boiler supplies to the engine 10 pounds of steam per pound of coal burned, is, - - -	116.5 mill. ft. lbs.
(*d*) The duty, per each million thermal units of heat imparted to the feed-water by the boilers neglecting steam used by feed-pumps, is, - - - - - - -	109.5 " " "
(*e*) Ditto, including steam used by pump, if latter exhausts into feed-water, - - -	109.3 " " "

Third.—The engine works smoothly while pumping against upward of 900 pounds pressure per square inch. Under fluctuations of steam pressure amounting to 5 per cent., and of oil pressure amounting to 10 per cent., the engine maintains an average length of stroke within three-thirty-seconds of an inch of a maximum possible stroke of 37.85 inches, or within 0.33 per cent. The greatest variation of individual strokes was nine-thirty-seconds of an inch, or 1.0 per cent.

Fourth.—The steam condensed in the jackets and reheater, in per cent. of the total steam supplied to engine, is 15.8 per cent.

Fifth.—The power wasted in friction of the mechanism, and operating the two air pumps, in per cent. of the horse-power of the steam end, is 3.6 per cent.

Sixth.—The actual delivery of oil was within 1.36 per cent. of the cubical displacement of the pump plungers, or, the "slip," was 1.36 per cent.

Seventh.—Without the use of steam in the jackets and reheater, water accumulates in the steam cylinders, in an hour's time, to such an extent as to be heard to splash against the pistons, while the loss of mean effective power, principally in the low cylinders, causes at least 10 per cent. loss of speed, which is probably attended with a considerable loss of economy, but to what extent was not determined. The use of steam jackets is undoubtedly necessary to the successful operation of a steam engine of this particular type.

APPENDIX.

MEAN EFFECTIVE PRESSURE OF STEAM CARDS. TEST OF APRIL 6TH, 1891.

No.	LOW CYLINDER. No. 1 Engine, Steam End.			LOW CYLINDER. No. 1 Engine, Pump End.			HIGH CYLINDER. No. 1 Engine, Steam End.			HIGH CYLINDER. No. 1 Engine, Pump End.		
	M. E. P. lbs. per sq. in.	Residuals.	Squares of residuals.	M. E. P. lbs. per sq. in.	Residuals.	Squares of residuals.	M. E. P. lbs. per sq. in.	Residuals.	Squares of residuals.	M. E. P. lbs. per sq. in.	Residuals.	Squares of residuals.
1	9.39	0.50	0.2500	9.30	0.12	0.0144	35.72	0.57	0.3249	33.00	0.73	0.5329
2	9.02	0.13	0.0169	9.36	0.18	0.0324	35.90	0.39	0.1521	32.90	0.63	0.3969
3	8.14	0.75	0.5625	8.48	0.70	0.4900	35.60	0.69	0.4761	32.65	0.38	0.1444
4	9.09	0.20	0.0400	8.30	0.88	0.7744	35.40	0.89	0.7921	31.05	1.22	1.4884
5	8.43	0.46	0.2116	8.41	0.77	0.5929	35.95	0.34	0.1156	32.20	0.07	0.0049
6	8.68	0.21	0.0441	8.36	0.82	0.6724	37.35	1.06	1.1236	31.30	0.97	0.9409
7	8.41	0.48	0.2304	8.30	0.88	0.7744	35.00	1.29	1.6641	33.35	1.08	1.1664
8	8.19	0.70	0.4900	8.81	0.37	0.1369	37.45	1.16	1.3456	33.85	1.58	2.4964
9	8.65	0.24	0.0576	8.62	0.56	0.3136	35.40	0.89	0.7921	32.65	0.38	0.1444
10	7.93	0.96	0.9216	8.52	0.66	0.4356	36.04	0.25	0.0625	31.90	0.37	0.1369
11	8.12	0.77	0.5929	8.26	0.92	0.8464	37.40	1.11	1.2321	31.90	0.37	0.1369
12	7.87	1.02	1.0404	8.53	0.65	0.4225	37.30	1.01	1.0201	31.50	0.77	0.5929
13	9.76	0.87	0.7569	9.69	0.51	0.2601	35.60	0.69	0.4761	32.70	0.43	0.1849
14	8.48	0.41	0.1681	8.93	0.25	0.0625	36.30	0.01	0.0001	31.40	0.87	0.7569
15	8.25	0.64	0.4096	8.72	0.46	0.2116	38.50	2.21	4.8841	31.15	1.12	1.2544
16	8.33	0.56	0.3136	8.35	0.83	0.6889	37.45	1.16	1.3456	31.25	1.02	1.0404
17	8.45	0.44	0.1936	9.00	0.18	0.0324	37.75	1.46	2.1316	31.65	0.62	0.3844
18	8.41	0.48	0.2304	8.05	1.13	1.2769	37.10	0.81	0.6561	31.10	1.17	1.3689
19	8.11	0.78	0.6084	8.42	0.76	0.5776	37.40	1.11	1.2321	31.90	0.37	0.1369
20	8.01	0.88	0.7744	8.33	0.85	0.7225	36.60	0.31	0.0961	32.80	0.53	0.2809
21	8.05	0.84	0.7056	8.08	1.10	1.2100	36.35	0.06	0.0036	32.30	0.03	0.0009
22	8.11	0.78	0.6084	8.28	0.90	0.8100	36.65	0.36	0.1296	30.20	0.07	4.2849

23	7.71	1.18	1.3924	8.34	0.84	0.7056	36.10	0.19	0.0361	32.40	0.13	0.0169
24	8.21	0.68	0.4624	8.51	0.67	0.4489	37.00	0.71	0.5041	32.45	0.18	0.0324
25	9.67	0.78	0.6084	10.09	0.91	0.8281	37.60	1.31	1.7161	33.00	0.73	0.5329
26	7.56	1.33	1.7689	8.60	0.58	0.3364	37.50	1.21	1.4641	32.30	0.03	0.0009
27	8.40	0.49	0.2401	8.75	0.43	0.1849	37.35	1.06	1.1236	32.10	0.17	0.0289
28	9.83	0.94	0.8836	9.78	0.60	0.3600	36.70	0.41	0.1681	32.65	0.38	0.1444
29	9.64	0.75	0.5625	10.13	0.95	0.9025	35.80	0.49	0.2401	32.80	0.53	0.2809
30	10.09	1.20	1.4400	10.40	0.22	0.0484	36.30	0.01	0.0001	33.10	0.83	0.6889
31	9.56	0.67	0.4489	10.13	0.95	0.9025	35.90	0.29	0.0841	32.50	0.23	0.0529
32	9.38	0.49	0.2401	9.50	0.32	0.1024	35.70	0.59	0.3481	32.70	0.43	0.1849
33	9.58	0.69	0.4761	9.99	0.81	0.6561	36.30	0.01	0.0001	32.65	0.38	0.1444
34	9.63	0.74	0.5476	10.01	0.83	0.6889	36.60	0.31	0.0961	32.65	0.38	0.1444
35	9.95	1.06	1.1236	10.03	0.85	0.7225	36.65	0.36	0.1296	32.80	0.53	0.2809
36	9.45	0.56	0.3136	9.82	0.64	0.4096	36.00	0.29	0.0841	32.20	0.07	0.0049
37	10.05	1.16	1.3456	10.02	0.84	0.7056	36.85	0.56	0.3136	32.85	0.58	0.3364
38	9.63	0.74	0.5476	9.92	0.74	0.5476	35.25	1.04	1.0816	32.25	0.02	0.0004
39	9.23	0.34	0.1156	9.75	0.57	0.3249	35.75	0.54	0.2916	32.30	0.03	0.0009
40	9.67	0.78	0.6084	9.86	0.68	0.4624	35.65	0.64	0.4096	32.40	0.13	0.0169
41	9.72	0.83	0.6889	10.27	1.09	1.1881	35.85	0.44	0.1936	32.85	0.58	0.3364
42	9.35	0.46	0.2116	9.14	0.04	0.0016	35.35	0.94	0.8826	32.40	0.13	0.0169
43	9.83	0.94	0.8836	10.10	0.92	0.8464	35.40	0.89	0.7921	32.40	0.13	0.0169
44	9.71	0.82	0.6724	10.00	0.82	0.6724	35.75	0.54	0.2916	32.90	0.63	0.3969
45	9.65	0.76	0.5776	10.05	0.87	0.7569	34.90	1.39	1.9321	32.45	0.18	0.0324
46	8.73	0.16	0.0256	9.25	0.07	0.0049	35.65	0.64	0.4096	31.65	0.62	0.3844
47	9.72	0.83	0.6889	10.17	0.99	0.9801	35.50	0.79	0.6241	31.35	0.92	0.8464
48	8.67	0.22	0.0484	9.31	0.13	0.0169	35.75	0.54	0.2916	31.85	0.42	0.1764
49	8.44	0.45	0.2025	9.29	0.11	0.0121	35.75	0.54	0.2916	32.20	0.07	0.0049
50	8.15	0.74	0.5476	9.11	0.07	0.0049	35.60	0.69	0.4761	32.40	0.13	0.0169
51	8.18	0.71	0.5041	8.60	0.58	0.3364						
Aver...8.89		Sum. of sq. 27.4035		Av. 9.19	Sum, of sq. 25.5165		Av. 36.29	Sum of sq. 34.3327		Av. 32.27	Sum. of sq. 22.9975	
0.079 probable error of average.				0.067 probable error of average			0.075 probable error of average			0.068 probable error of average		

MEAN EFFECTIVE PRESSURE OF STEAM CARDS TEST OF APRIL 16TH, 1891.

No.	LOW CYLINDER. No. 2 Engine, Steam End			LOW CYLINDER. No. 2 Engine, Pump End.			HIGH CYLINDER. No. 2 Engine, Steam End.			HIGH CYLINDER. No. 2 Engine, Pump End.		
	M. E. P. lbs. per sq. in.	Residuals.	Squares of residuals.	M. E. P. lbs. per sq. in.	Residuals.	Squares of residuals.	M. E. P lbs. per sq. in.	Residuals.	Squares of residuals.	M. E. P. lbs. per sq. in.	Residuals.	Squares of residuals.
1	9.00	0.11	0.0121	9.69	0.55	0.3025	37.60	1.28	1.6384	32.40	0.96	0.9216
2	9.12	0.23	0.0529	8.76	0.33	0.1089	38.25	1.93	3.7249	32.45	1.01	1.0201
3	9.75	0.86	0.7396	10.38	1.29	1.6641	38.30	1.98	3.9204	31.90	0.46	0.2116
4	7.70	1.19	1.4161	8.33	0.76	0.5776	39.10	2.78	7.7284	33.60	2.16	4.6656
5	8.21	0.68	0.4624	8.35	0.74	0.5476	37.55	1.23	1.5129	31.50	0.06	0.0036
6	8.17	0.72	0.5184	8.27	0.82	0.6724	38.40	2.08	4.3264	32.10	0.66	0.4356
7	8.27	0.62	0.3844	8.60	0.49	0.2401	37.60	1.28	1.6384	33.20	1.76	3.0976
8	8.03	0.86	0.7396	8.10	0.99	0.9801	37.50	1.18	1.3924	33.80	2.36	5.5696
9	7.97	0.92	0.8464	7.90	1.19	1.4161	36.70	0.38	0.1444	31.25	0.19	0.0361
10	8.29	0.60	0.3600	8.21	0.88	0.7744	36.90	0.58	0.3364	30.90	0.54	0.2916
11	8.28	0.61	0.3721	8.41	0.68	0.4624	37.25	0.93	0.8649	31.80	0.36	0.1296
12	9.73	0.84	0.7056	9.87	0.76	0.5776	37.10	0.78	0.6084	30.10	1.34	1.7956
13	8.76	0.13	0.0169	8.70	0.39	0.1521	35.50	0.82	0.6724	31.00	0.44	0.1936
14	7.94	0.95	0.9025	8.48	0.61	0.3721	35.80	0.52	0.2704	31.80	0.36	0.1296
15	8.09	0.80	0.6400	8.64	0.45	0.2025	35.60	0.72	0.5184	32.00	0.56	0.3136
16	8.19	0.70	0.4900	8.62	0.47	0.2209	37.40	1.08	1.1664	31.15	0.29	0.0841
17	8.08	0.81	0.6561	8.27	0.82	0.6724	36.40	0.08	0.0064	31.45	0.01	0.0001
18	8.39	0.50	0.2500	8.67	0.42	0.1764	36.10	0.22	0.0484	30.60	0.84	0.7056
19	8.03	0.86	0.7396	8.45	0.64	0.4096	36.30	0.02	0.0004	31.20	0.24	0.0576
20	7.74	1.15	1.3225	7.93	1.16	1.3456	36.30	0.02	0.0004	31.20	0.24	0.0576
21	8.36	0.53	0.2809	8.24	0.85	0.7225	36.65	0.33	0.1089	31.05	0.39	0.1521
22	7.89	1.00	1.0000	8.51	0.58	0.3364	36.70	0.38	0.1444	31.75	0.31	0.0961
23	8.20	0.69	0.4761	8.41	0.68	0.4624	36.90	0.58	0.3364	30.75	0.69	0.4761
24	8.33	0.56	0.3136	8.49	0.60	0.3600	36.45	0.13	0.0169	31.00	0.44	0.1936
25	8.85	0.04	0.0016	9.66	0.57	0.3249	36.80	0.48	0.2304	31.60	0.16	0.0256

26	9.64	0.75	0.5625	9.83	0.74	0.5476	38.15	1.83	3.3489	33.60	2.16	4.6656
27	9.49	0.60	0.3600	9.56	0.47	0.2209	36.00	0.32	0.1024	30.50	0.94	0.8836
28	9.54	0.65	0.4225	9.62	0.53	0.2809	35.50	0.82	0.6724	31.30	0.14	0.0196
29	9.73	0.84	0.7056	9.78	0.69	0.4761	35.40	0.92	0.8464	30.60	0.84	0.7056
30	9.73	0.84	0.7056	9.70	0.61	0.3721	35.30	1.02	1.0404	31.10	0.34	0.1156
31	9.82	0.93	0.8649	9.91	0.82	0.6724	35.10	1.22	1.4884	30.80	0.64	0.4096
32	9.56	0.67	0.4489	9.62	0.53	0.2809	35.05	1.27	1.6129	31.05	0.39	0.1521
33	9.26	0.37	0.1369	9.37	0.28	0.0784	34.75	1.57	2.4649	31.35	0.09	0.0081
34	9.61	0.72	0.5184	9.65	0.56	0.3136	35.45	0.87	0.7569	31.35	0.09	0.0081
35	8.31	0.58	0.3364	8.81	0.28	0.0784	35.50	0.82	0.6724	31.35	0.09	0.0081
36	9.55	0.66	0.4356	9.56	0.47	0.2209	35.45	0.87	0.7569	31.30	0.14	0.0196
37	9.53	0.64	0.4096	9.47	0.38	0.1444	34.30	2.08	4.3264	31.40	0.04	0.0016
38	9.10	0.21	0.0441	9.50	0.41	0.1681	34.75	1.57	2.4649	30.25	1.19	1.4161
39	9 72	0.83	0.6889	9.71	0.62	0.3844	35.40	0.92	0.8464	30.85	0.59	0.3481
40	9.90	1.01	1.0201	10.10	1.01	1.2001	37.00	0.68	0.4624	30.65	0.79	0.6241
41	10.00	1.11	1.2321	9.98	0.89	0.7921	35.60	0.78	0.6084	31.55	0.11	0.0121
42	10.01	1.12	1.2544	9.90	0.81	0.6561	35.65	0.67	0.4489	32.35	0.91	0.8281
43	9.80	0.91	0.8281	10.13	1.04	1.0816	34.70	1.62	2.6244	31.40	0.04	0.0016
44	9.26	0.37	0.1369	9.17	0.08	0.0064	35.90	0.42	0.1764	31.60	0.16	0.0256
45	9.11	0.22	0.0484	9.63	0.54	0.2916	34.75	1.57	2.4649	30.85	0.59	0.3481
46	9.50	0.61	0.3721	9.68	0.59	0.3481	35.90	0.42	0.1764	30.65	0.79	0.6241
47	....	....	....	8.80	0.29	0.0841	36.15	0.17	0.0289	31.10	0.34	0.1156
48	....	....	....	....	....	....	35.15	1.17	1.3689	30.75	0.69	0.4761
49	....	....	....	....	....	....	36.45	0.13	0.0169	31.65	0.21	0.0441
50	....	....	....	....	....	....	36.70	0.38	0.1444	31.15	0.29	0.0841
51	....	....	....	....	....	....	36.40	0.08	0.0064	30.75	0.69	0.4761
52	....	....	....	....	....	....	36.55	0.23	0.0529	31.10	0.34	0.1156
Aver..8.89		Sum of sq., 25.2214		Av. 9.09	Sum of sq., 22.6007		Av. 36.32	Sum of sq., 61.3263		Av. 31.44	Sum of sq., 33.2012	
0.074 probable error of average.				0.068 probable error of average			0.102 probable error of average			0 075 probable error of average		

Aggregate error reduced to area of low cylinder.

$$\left.\right\} = \sqrt{\left(\overline{0.079}^2 + \overline{0.067}^2 + \overline{0.074}^2 + \overline{0.068}^2 + \frac{\overline{0.075}^2 + \overline{0.068}^2 + \overline{0.102}^2 + \overline{0.075}^2}{[4.097]^2}\right)} = \left\{\begin{array}{l}\text{0.17 lbs. per sq. in.,}\\ \text{or 1\% of steam H. P.}\end{array}\right.$$

NASHVILLE TENN.

10,000,000 GALLONS CAPACITY.

ONE ENGINE.

NASHVILLE, TENN., December 1, 1891.

To the Hon. Board of Public Works and Affairs:

GENTLEMEN—I have the honor to submit herewith my report of the duty and capacity test of the new pumping engine, recently erected by Henry R. Worthington, at the new pumping station.

The contract requires an engine "which shall be easily capable of pumping from low water in the well at pumping station into the reservoir on Kirkpatrick's Hill, through the 36-inch main, already laid, ten million U. S. gallons of water each 24 hours."

The guaranteed duty is eighty million foot-pounds, "and shall be based on an evaporation of eight hundred pounds of feed-water into steam at eighty pounds boiler pressure, the feed-water to be furnished by an independent feed-pump and boiler, and all the feed-water will be charged to engine, allowance being made for water entrained in steam, as found by calorimeter tests. Allowance will also be made for any additional heat imparted to feed-water, the temperature of river water being taken as the normal temperature of feed-water.

"In estimating the duty, the quantity of water pumped shall be determined by reservoir measurement, and the head against which pumps work will be the dynamic head, as shown by gauge on discharge main, plus the distance from said gauge to the mean level of water in the wet well from which water is taken, plus one pound friction of water through pumps. The weight of water will be taken at sixty-two and one-half (62½) pounds per cubic foot."

DESCRIPTION OF ENGINE.

The engine is of the Worthington high duty vertical type, and has two high-pressure cylinders placed above and in line with two low-pressure cylinders, which latter are supported by a bed plate spanning a solid masonry well. There is a surface condenser located in the delivery main, through which the exhaust steam is passed, and the condensed steam is removed by an independent air pump.

The sides and heads of the high and low-pressure cylinders are jacketed with steam at boiler pressure, and the water of condensation drains into a closed tank, and is returned to boilers, without loss of pressure, by a small, independent steam pump, controlled by a float.

Reheating tubes are placed in the high-pressure exhaust pipe, and the steam passes through a separator located just below the main throttle valve.

The Worthington High Duty attachment, which enables the use of steam expansively, has already become so well known in this country that it scarcely needs a detailed description. Suffice it is to say that this one is of the usual form, consisting of two compensating cylinders, acting on each rod just below the low-pressure cylinders, the pressure in these compensating cylinders being taken from the water main after being intensified by a differential piston or accumulator.

DESCRIPTION OF PUMPS.

Two double-acting plunger pumps are placed at the bottom of the 50-foot dry well, one being in line with each pair of steam cylinders, and the plungers are coupled to the same rod which passes through high and low-pressure steam cylinders.

The plungers work in and out of each water cylinder through stuffing boxes, and the water valves are composed of phosphor bronze disks and are closed by springs.

BALANCING DEVICE.

The weight of the moving parts on each side is carried by a single-acting plunger attached to the piston rod, and working in a cylinder supported from the under side of the bed plate. These balancing cylinders contain water and are connected by a 14-inch cast-iron pipe to a large air tank, which serves to keep the balancing pressure very nearly constant.

METHOD OF CONDUCTING TEST.

The contract requires the measurement of water to be made in the reservoir, but as this would have necessitated the reservoir being shut off for several days, after advising with your Honorable Board, it was agreed to measure the water by displacement. As the plungers are outside packed no leakage could occur there without being seen, and in order to find the leakage of the valves a test was made by closing a gate on the discharge main and measuring the quantity of water required to maintain the pressure by a meter. Each set of valves was tested separately, and the leakage was thus shown to be eleven and seven-tenths ($11\frac{7}{10}$) gallons per minute, which is $\frac{15}{100}$ of 1 per cent. of the total water pumped by the engine. As, however, the leakage of valves on Holly pumps was found by reservoir measurement to be $1\frac{84}{100}$ per cent., it was agreed to assume this amount in calculating the duty.

For convenience of measurement cold water was fed to the boilers, and the heat discharged by the jacket drains and exhausts of small pumps was noted and the engine credited with the same.

The feed-water was weighed in one tank, and afterwards run to another tank, from which it was pumped to the boiler.

Steam traps were placed on the separator drain, and also on the main steam pipe between the separator and the boilers. The water from these traps was discharged into a barrel placed in a platform scales, and weight of water thus collected was also credited to the engine. These traps did not work during the first part of the test.

A barrel calorimeter was attached to steam pipe between separator and engine.

The average length of several strokes on each side were noted every 15 minutes.

During the first 24 hours the cushion valves were frequently adjusted, but for the last 24 hours they were only occasionally touched. The length of the stroke during the first 24 hours was $61\frac{96}{100}$ inches. For the last 24 hours it was $61\frac{91}{100}$. This shows the stroke to be remarkably even. The difference between the first part of the test, when it was tried to keep the stroke as long as possible, and the latter part, when little or no attention was paid to it, being insignificant.

In order to avoid all question the stroke was assumed as shorter than actually observed, and was taken at $61\frac{75}{100}$ inches.

At about three o'clock A. M., May 14th, a joint was blown out of the feed pump and considerable water was lost, no allowance for which has been made. The head against which the pumps were to work was to include the friction due to the flow of 25,000.000 gallons through the pipe line each 24 hours. To obtain this additional head, the gate on the discharge main was partially closed.

DIMENSIONS.

Diameter of each high-pressure cylinder (two),	41	ins.
Diameter of each high-pressure piston-rod,	7.25	"
Diameter of each low-pressure cylinder (two),	82	"
Diameter of each low-pressure piston-rod,	7.75	"
Diameter of each plunger (two double acting),	26.5	"
Diameter of each plunger-rod,	7.75	"
Nominal stroke,	60	"
Contact stroke,	62.1	"

DATA RELATING TO WORK OF PUMP.

Area of plunger,	551.54	sq. ins.
One-half area of rod,	23.58	"
Effective area of plunger,	527.96	"
Average length of stroke,	61.75	ins.
Counter at beginning of trial,	32,978	
Counter at end of trial,	71,382	
Total number of counts,	38,404	
Number of strokes per count,	4	
Total number of single strokes,	153,616	
Plunger travel during trial,	790,482.33	ft.
Displacement per foot of plunger travel in pounds of water,	229.149375	
Total water pumped during trial,	181,138,528	lbs.
Deduction for slip, $1\frac{84}{100}$,	3,332,948	"
Net pounds of water pumped,	177,805,580	"
Weight of 1 U. S. gallon of water at 70° Fahr.,	8.333	"
U. S. gallons of water pumped during trial (48 hours), deducting slip,	21,319,613	
U. S. gallons pumped in 24 hours,	10,659,806	
Pressure by gauge on force main, pounds	127.15	
Equivalent head in feet,	293.71	
Vertical distance from water in pump well to centre of gauge,	66.48	ft.
Contract allowance for friction in pump,	2.31	"
Total head in feet,	362.5	"

DATA RELATING TO WORK OF STEAM CYLINDERS.

Total weight of water furnished to boilers,	607,809	lbs.
Weight of water trapped from steam pipe,	4,123	"
Weight of steam in calorimeter,	286	"

Weight of steam equal to heat returned through hot well,	39,448 lbs.
Moisture in steam by calorimeter,	4.25 per cent.
Net dry steam used by engine,	539,084 lbs.
Steam pressure by gauge,	82.65 "
Vacuum in inches of mercury,	27.39
Average area of high-pressure piston,	1,298.1
" " low " "	5,236.8 sq. in.
" M. E. P. on H. P. piston,	36.1 lbs.
" " " " L. P. "	8.5 "
" piston travel per minute (both sides),	274.4 ft.
" I. H. P. steam end,	764.6
" H. P. water column,	697.2
Mechanical efficiency, $\frac{697.2}{764.6}$	91.2 per cent.
Dry steam per indicated horse-power per hour,	14.6 lbs.
Heat units consumed, per I. H. P. per hour,	17,050.

HEAT RETURNED THROUGH HOT WELL.

Meter at end of test,	15,424
" " beginning of test,	7,653
Cubic feet water registered,	7,771
Average temperature of water,	165.28 degs.
" " " river water,	70 "
Rise of temperature,	95.28 "
Weight of 1 cubic foot water at 165.28 deg.,	60.87 lbs.
Weight of water through meter,	473,020 "
Heat units in water, 473,020x95.28,	45,069,345
Equivalent pounds of steam at 80 pounds gauge pressure,	39,448

GENERAL DATA.

Test began at 11 A. M., May 12, 1891. Test ended, 11 A. M., May 14, 1891.

Duration of test,	48 hours.
(*a*) Pounds of water pumped (by displacement),	181,138,528
(*b*) Net water pumped (deducting slip),	177,805,580
(*c*) Head against which pumps worked,	362.5
(*d*) Dry steam furnished at engine,	563,952 lbs.
(*e*) U. S. gallons water pumped during trial,	21,319,613 gals.
(*f*) Heat units in dry steam furnished at engine,	616,380,828

RESULTS.

		Deducting 1.84% Slip,	Plunger Displacement.
Duty on 8 lbs. Evaporation,	$\frac{b\times c\times 800}{d}$ -	95,650,436	97,104,911
" " 10 " "	$\frac{a\times e\times 800}{d}$ -	119,563,040	121.762,999
" " 1,000,000 heat units,	$\frac{b\times c\times 1,000,000}{f}$	104.569.317	106,495,000
Capacity, $\frac{e}{2}$ - - - - - - - -		10,659.806	

From the above you will see that the engine has far exceeded the contract requirements.

The duty obtained, as required by the contract, was 95,650,430, while the guarantee duty was 80,000,000.

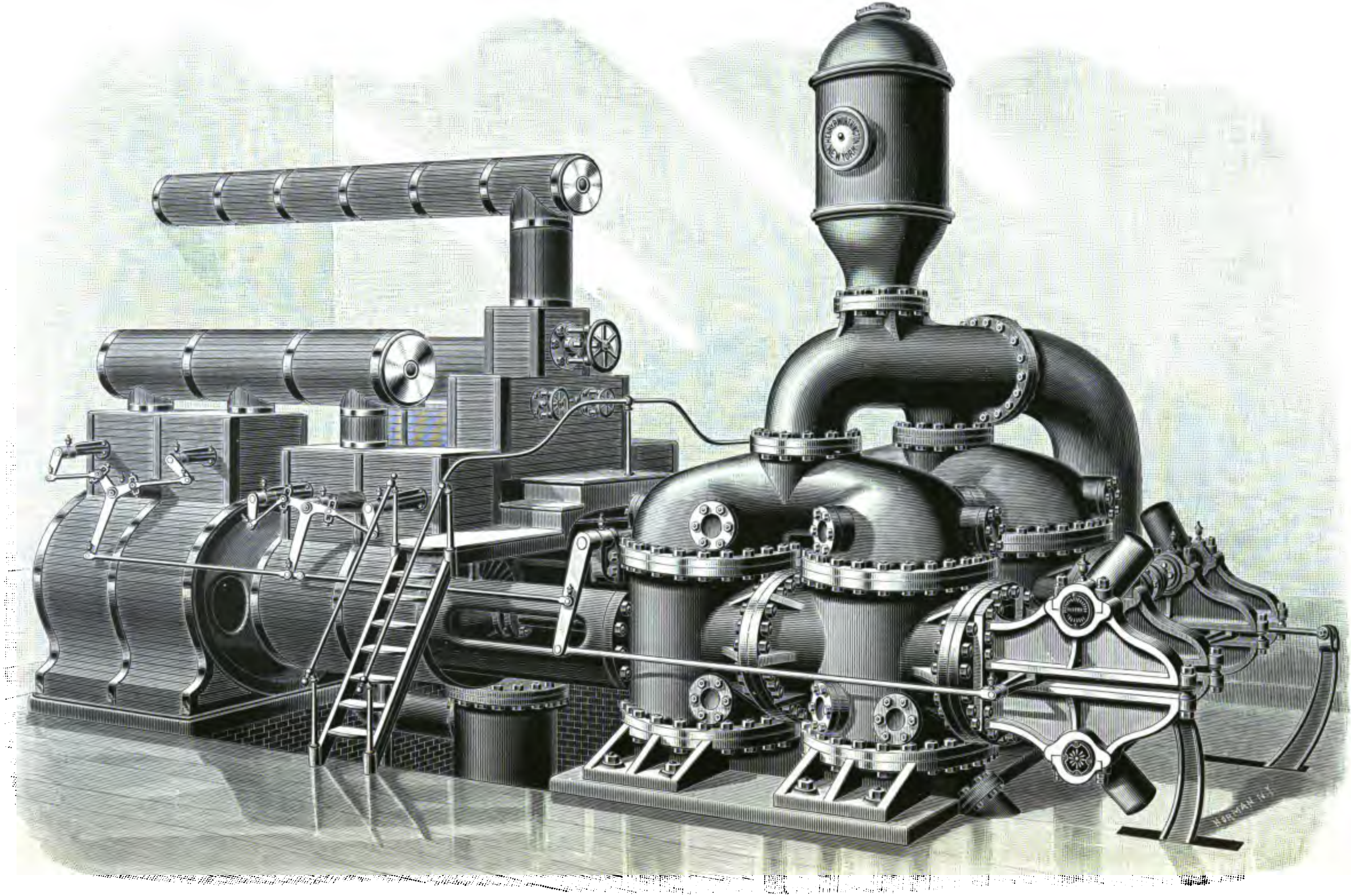
The quantity of water pumped in 24 hours was 10,659,806 gallons. The quantity guaranteed was 10,000,000.

The engine has been in service almost continually since the test was made and has done excellent work.

Respectfully submitted,

GEO. REYER,

Supt. Water Works.



WORTHINGTON HIGH DUTY PUMPING ENGINE AT LOWELL, MASS.

LOWELL, MASS.

10,000,000 GALLONS CAPACITY.

ONE ENGINE.

REPORT OF GEORGE H. BARRUS, M. E.*

OFFICE OF GEO. H. BARRUS,
95 MILK STREET, ROOM 45,
BOSTON, February 23, 1892.

Lowell Water Board, Lowell, Mass.:

GENTLEMEN—Acting under your instructions, I conducted the tests of the new ten-million gallon High-Duty Worthington Pumping Engine, as called for by the contract, on February 11th and 12th, 1892, and I beg to report upon the same as follows:

THE STIPULATIONS OF THE CONTRACT.

The stipulations of the contract relating to the performance of the engine are to the effect that, in the matter ot capacity, the engine shall be capable of pumping at the rate of 10,000,000 gallons of water in 24 hours into a 30-inch force main leading from the engine house to the reservoir, and it shall be capable of twenty (20) per cent. greater capacity—that is, 12,000,000 gallons of water in 24 hours, without danger of injury to the working parts. It is also stipulated that the slip shall not be in excess of four (4) per cent. In the matter of duty, it is stipulated that the engine shall be capable of performing a duty of 110,000.000 foot-pounds of work on a consumption of 1,000,000 heat units; and, further, that the method of test shall be in accordance with that determined upon by the committee of the American Society of Mechanical Engineers for devising a standard method of conducting duty trials. (See Vol. XII., page 530, Transactions, A. S. M. E., 1890.)

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* Published by the Lowell Water Board, 1892.

DESCRIPTION OF THE ENGINE.

The engine is the latest pattern of the Worthington High-Duty Pumping Engine, having cylinders arranged on the compound, direct-acting principle. The general features of construction are shown in the accompanying cuts, one of which is a longitudinal section, showing the general arrangement of the principal parts; and the other, an outside view of an engine of the same design. These cuts, which have been kindly furnished by the manufacturers of the engine, will give to those not already familiar with it a general idea of its construction, which will be sufficient for the purposes of this report.

The steam cylinders are all jacketed, both on the sides and heads, with steam of boiler pressure, and reheaters are provided, through which the steam passes on its way from the high to the low cylinders, which are likewise steam jacketed. The steam which is used in the jackets is that derived from the drain pipe of the separator which belongs to the engine, and the steam from this point passes in succession through the reheaters, which are at the highest elevation, and thence through the jackets of the four cylinders, finally being delivered into a common drain pipe, which proceeds to a tank in which the water of condensation is collected. The jacket tank is drained by a small duplex steam pump working automatically, the throttle valve being under the control of a float in the tank, and this water is pumped into the boilers. The size of the pump is 3x2x3 inches. The pipe leading from the pump to the boilers is independent of any other feed pipe. The exhaust steam discharged from this pump is carried to an iron tank in the basement of the engine-house, which serves as a hot well for the supply of water to the boilers, being itself supplied by the overflow water discharged from the air pumps of the engine. The hot well tank is one which has formerly served the same purpose for the old engines in the pumping station. Its diameter is 30 inches, and its length 20 feet.

In addition to the jacket pump, there is a second steam pump connected with the engine, which, by means of a set of pistons, furnishes air to the top of the accumulator cylinder, and, by means of a set of plungers, furnishes water to the under side of the accumulator ram. The size of the pump, which is of the duplex type, is 4½x3x4 inches. The exhaust steam from this pump, like that from the jacket pump, is carried to the hot well tank just referred to.

The air pumps of the condensing apparatus are operated by direct connections with the main engine. The only accessories about the engine itself which consume steam are the jackets and the two direct-acting steam pumps referred to.

The feed pump, for supplying the boilers with water, is one which has already been in use for the old plant. It is located in the cellar, under the engine-room, near by the hot well tank. The exhaust steam from this pump is carried into the tank, and a branch pipe enables it to be used when desired for heating the radiators in the engine-room. Its size is 6x4x6 inches, duplex.

Besides the separator, which is incorporated with the engine, there is an additional separator of the "Stratton" type, attached to the steam pipe near by. This apparatus is intended, eventually, to be drained into the jacket tank.

The cylinders are protected from radiation by a covering of magnesia, outside of which is a layer of hair felting, and the whole is encased in black walnut lagging. The steam pipe, extending the whole distance from the boilers to the engine, is covered with magnesia, and that part within the engine-room is finished with black walnut lagging.

The principal dimensions of the engine are given in the following table:

TABLE NO. 1.—DIMENSIONS OF ENGINE.

Number of high-pressure cylinders, - - - -	2
Number of low-pressure cylinders, - - - -	2
Number of water cylinders, - - - - - -	2
Diameter of high-pressure cylinders, - - - -	25 ins.
Diameter of low-pressure cylinders, - - - -	50 "
Diameter of water plungers, - - - - - -	27½ "
Length of stroke of all cylinders, maximum, - -	38 ins.
Diameter of high-pressure piston-rod, front, - -	5 "
Diameter of high-pressure piston-rod, back, - -	5¼ "
Diameter of low-pressure piston-rod, one end, - -	5 "
Diameter of piston-rod of water cylinder, front end, -	5¼ "
Diameter of piston-rod of water cylinder, back end, -	$3\frac{7}{16}$ "
Clearance of high-pressure cylinder, minimum, - -	2 per ct.
Clearance of low-pressure cylinder, minimum, - -	$1\frac{5}{10}$ "
Ratio of volume of low-pressure cylinder to that of the high-pressure cylinder, - - - -	4.15 to 1
Diameter of steam pipe, - - - - - - -	6 ins.
Length of steam pipe from boilers to engine, - -	146 ft.
Volume of pump cylinder corresponding to one displacement at full stroke of 38 inches, - -	95.16 gals.

Diameter of suction main, - - - - - - -	30 ins.
Diameter of force main at engine-house, - - -	30 “
Diameter of force main now supplying water to reservoir, - - - - - - - - -	24 “
Elevation of crest of the weir at the reservoir above the engine-room floor, - - - - - -	143.5 ft.
Elevation of crest of the weir at the reservoir above the average water line in the pump well during the test of February 12th, 1892, - - -	155.5 “

The water is discharged by the engine through a 30-inch force main, which first drops into the cellar, where it contains a check valve and gate, and thence passes horizontally out of the building for a distance of about 40 feet. Here it terminates in a Y, one outlet of the same, which is eventually to be connected to the reservoir, being plugged. The branch from the Y leads to the 24-inch force main, which has, up to the present time, been in use in connection with the old engines. The water is discharged at the reservoir, first, into a stone chamber, one side of which is provided with a weir. The water, on leaving the weir, empties into a second chamber, which communicates with the reservoir. By the side of these two chambers, which are termed the “inlet” chambers, there is an “outlet” chamber, which receives the water from the reservoir, and conducts it to the distributing pipe leading to the city service. Between the inlet and the outlet chambers there is an opening fitted with a sluice gate, for carrying the water, when desired, directly from the inlet chamber into the distributing pipe without first passing through the reservoir. The location of the weir, and the general features of the inlet chamber, are shown in the sketches given on page 14. The pipes leading into the 24-inch main from the two old engines are provided with gates, and that which leads from the old Worthington Engine is also provided with a check valve. At a point on the 24-inch main, at a distance of about 500 feet from the engine-house, there is a 12-inch branch which connects the force main to the distributing pipes of the city. This branch is fitted with a gate valve. At the lowest point in the force main, between the reservoir and and the engine-house, there is a blow-off valve attached to the bottom of the pipe, by means of which the whole length of the force main can be emptied. At a distance of about 1,000 feet the 24-inch main is fitted with a check valve.

The steam which supplies the new engine is furnished by new boilers, which were introduced especially for this purpose. They are of the horizontal return tubular type and two in number. They are located in the boiler-room, at the end of the building farthest from the engine and nearest the chimney. The steam is taken from the forward end of each shell through a 5-inch vertical pipe. The two pipes are connected into a 6-inch pipe, which extends horizontally to the rear end of the boilers, where it rises vertically to the long run of horizontal pipe which leads to the engine. The boilers are set in brickwork, in the ordinary way, with side walls having air spaces. There is no provision for admitting air above the fuel, save that which gains entrance through the fire-doors, which are provided with the usual registers. The grates are of the shaking pattern, with about 40 per cent. air space. The flue gases on leaving the smoke arches in front, the shells being of the overhanging pattern, pass into a flue built of brickwork, and located above and between the boilers, thence to the rear of the setting, and, finally, descend into the underground flue leading to the chimney by means of a brick drop flue. The flue above the boilers contains a wrought-iron shell without tubes, 20 inches in diameter and 16 feet long, through which the feed water is pumped on its way to the boilers, and this apparatus serves as a flue heater. The damper by which the draft is regulated is placed in the drop flue, and operated by one of Locke's automatic damper regulators.

The principal dimensions of the boilers are as follows:

TABLE NO. 2.—DIMENSIONS OF BOILERS.

1.	Number of boilers,	2
2.	Diameter of each shell,	72 ins.
3.	Length of shell between heads,	16 ft.
4.	Number of tubes in each boiler, 3 ins. outside diameter,	140
5.	Size of gates,	6 ft. by 6 ft.
6.	Area of heating surface in each boiler,	1,755 sq. ft.
7.	Area of grate surface in each boiler,	36 "
8.	Area for draft through tubes in each boiler,	5.7 "
9.	Ratio of heating surface to grate surface,	48.8 to 1
10	Ratio of grate surface to tube area,	6.5 to 1
11.	Area through main flue,	11.1 sq. ft.
12.	Height of chimney,	100 ft.
13.	Distance of boilers to engine measured along the main steam pipe,	146 ft.

METHOD OF CONDUCTING THE TESTS.

The method followed in conducting the trial for duty and capacity was that of the American Society of Mechanical Engineers, as prescribed in the contract. In determining the amount of duty performed, the principal data required are the weight of water supplied to the boilers by the main feed pump, and by the jacket pump, the temperature of the water supplied from these two sources, the boiler pressure, from a knowledge of which the total heat of the steam is obtained, the quantity of water drained from the separator, the quantity of moisture in the steam supplied to the engine, the pressures in the force main and suction main close to the engine, and the quantity of water pumped deduced from the plunger displacement. The quantity of water actually discharged at the reservoir was made the subject of weir measurement, using the weir permanently located at the entrance of the force main to the reservoir.

The method of conducting the tests will be considered under three headings: the preliminary run, for determining the working temperatures; the main test, for determining the weight of water supplied to the boiler and the various other quantities; and the weir test.

THE PRELIMINARY TEST FOR DETERMINING WORKING TEMPERATURES.

This test was made February 11th, 1892. Since the contractor furnished simply the engine, and did not furnish either the boilers or the main feed pump, and since the Standard Method allows the contractor to specify the kind of pump which shall be used for feeding during the test, the engine is entitled to all the heat exhausted from this pump, as well as that rejected by the engine itself and its accessories. The rejected heat determines the temperature of the water supplied to the boiler by the main feed pump, and it was this temperature which was ascertained by the preliminary trial. The feed water for the purpose was drawn from a small hot well temporarily applied, which received water from the overflow of the condenser, and into this well the exhaust steam of the three pumps was led and condensed. The temperature of the water was taken as it left the hot well and entered the pump. During the preliminary trial the engine was worked at its rated capacity of 10,000,000 gallons in twenty-four hours, and at a boiler pressure of 92 pounds per square inch. No measurements were taken of the quantity of feed water supplied by either the main feed pump or the jacket pump, the plant being operated under working conditions, with the single exception that the separator water was allowed to run to waste. The feed pump was worked at such a speed as to just keep up the supply of water in the boilers, and no more. Under these conditions it was found that the temperature of the water leading to the main pump was maintained at 175 degrees. The temperature of the water supplied by the jacket pump was 322 degrees. It was found on this test that when the pump was worked at a speed corresponding to that which was maintained during the main trial of the engine, when the capacity was somewhat in excess of its rating, that the temperature of the feed water dropped to 170 degrees. Consequently, this latter temperature is the one which has been adopted as the working temperature of the main feed water for the purposes of the duty trial. The temperature of the jacket water under the conditions of the main trial, the boiler pressure being carried at a higher point than on the preliminary trial, was 326 degrees, and this temperature is the one on which the results of the main trial are based.

THE MAIN TRIAL.

The main trial was made February 12th, 1892. It followed closely the plan laid down by the Standard Method, as given on page 546 of Vol. XI., Transactions, A. S. M. E. It embraced a complete test of the plant, not only of the engine, but of the new boilers as well. To measure the main feed water and the jacket water it was necessary to operate the plant under slightly different conditions from the working conditions employed on the preliminary run. The feed water was all supplied by the main feed pump, and this was taken from the large hot well tank in the cellar, passing through the weighing tanks before its supply to the pump. The jacket water was drawn off into two barrels, in which it was weighed, and afterwards thrown away. To make the conditions as nearly as possible the same as the usual working conditions, the jacket pump was kept in operation at its ordinary speed, though not engaged in doing its full quantity of work, the plungers having been removed. The separator water was allowed to go to waste, after being weighed, in the same manner as on the preliminary trial.

The new fires for the boiler test were started with wood at 7.35 A. M. The engine was soon set to work, and thereafter the regular observations of the instruments concerned in the trial were begun. The engine was kept in operation until 9.25 P. M., in all nearly 14 hours, and the fires, which had previously been burned down, were dropped at 9.43 P. M. The period selected for the engine test, or duty trial proper, was that beginning at 10 A. M. and ending at 8 P. M., the duration of the period being exactly 10 hours.

The instruments observed during the progress of the trial are those referred to under appropriate headings in the following description, which gives an account of the data, and the manner in which it was determined.

1. FUEL —The wood used in lighting the fires, the coal, and the ashes, were weighed on the scales in daily use in the boiler-room. A sample of each barrow load of coal was laid by, and subjected to a heat test in the writer's Coal Calorimeter. A sample of the ashes at the end of the test was selected, and sifted in a screen having ⅜-inch meshes, thereby determining the proportion of unconsumed coal in the refuse, and the proper deduction has been made from the total weight of coal fired for the unconsumed coal thus found. The fires were carried at a depth of about 6 inches, and the firing was done on the spreading system. The coal was dry and fired in its dry state.

2. DRAFT.—The amount of draft acting upon the boilers inside of the regulating damper was observed every fifteen minutes during the trial. The instrument used was the writer's Differential Draft Gauge, which multiplies the ordinary suction of the draft eight times.

3. FLUE TEMPERATURE.—The temperature of the flue was taken every fifteen minutes from a high grade thermometer set in the top of the main flue, close to the two uptakes. This thermometer at the conclusion of the trial was found to be broken, and the record of the test in this particular is based on observations taken on a subsequent run.

4. FEED WATER.—The weighing apparatus for the main feed water consisted of two tanks, one placed on scales above the other. The water was first drawn into the upper tank, which held a charge of about 460 pounds, and, after being weighed, it was emptied into the tank below. The suction pipe of the feed pump was connected into the lower tank. Each tank full of water was weighed, the time of each emptying noted, and the temperature of the water in the receptacle below observed for each charge. In connection with the measurements of feed water, the height of water shown in the gauge glasses on the boilers was observed every fifteen minutes.

5. JACKET AND SEPARATOR WATER.—The jacket water was measured in two barrels, each of which was placed on scales, and

the water was drawn into them alternately, one being filled while the other was emptying. Each barrel was supplied at the start with a sufficient quantity of cold water to prevent evaporation from the highly heated liquid drawn from the jackets. The time was noted when each change was made from one barrel to the other, and the temperature of the water in the discharge pipe of the pump was observed at the same time. The level of the water in the jacket tank, as shown in the gauge glass attached to it, was maintained as nearly as possible at a uniform level. The separator water was allowed to drain into a tub placed on scales, and observations of its weight were taken every hour. The quantity of water derived from this source was so small that one filling of the tub sufficed for the whole run. This tub was also provided at the start with cold water, to prevent evaporation from the highly heated separator water. The level of water in the separator, as shown in the gauge glass attached to it, was maintained at a uniform height.

6. MOISTURE IN STEAM.—The moisture in the steam was determined by the use of the heat gauge of one of the writer's Universal Calorimeters, which was attached between the "Stratton" Separator and the Worthington Separator, the point of attachment being at the side of the elbow just above the latter. The steam discharged from the calorimeter was carried into the cellar, and the quantity thus taken away from the engine was determined by condensing it for a short trial in a pail of water. The quantity used was found to be at the rate of 39 pounds per hour. The readings of the two thermometers cf the calorimeter were taken every fifteen minutes. It may be remarked here that they showed almost constant indications throughout the trial.

7. GAUGES.—The steam and vacuum gauges used were those belonging to the engine, and attached to the marble slab. The steam gauge which is attached to the main steam pipe above the engine was found to register a pressure 2 pounds above the actual. These were observed every fifteen minutes. The water gauge attached to the force main, and the vacuum gauge attached to the suction main, were test gauges belonging to the writer. These were read every fifteen minutes. The former was attached to an air chamber, as described in the report of the Standard Method, which was connected to the engine with the same pipe as the water gauge belonging to the engine. The valves of both gauges were wide open, and both gauges indicated a steady pressure with practically no vibration of the pointer. The distance from the

water line in the air chamber noted, to the center of the vacuum gauge on the suction pipe, was 3.91 feet. The gauge pipe on the force main was attached to the chamber above the discharge valves of the pump, furthest from the Morris Engine, at the end nearest the steam cylinders. The vacuum gauge on the suction main was attached to a pipe leading to the four suction chambers of the engine beneath the suction valves. The pointer of this gauge showed considerable vibration, and its valve was throttled, so as to largely overcome it. Attached to the same pipe was a mercury column, which was read at the same time as the vacuum gauge, and found to agree therewith. The gauge on the force main was verified at the beginning of the trial by comparison with a dead weight testing apparatus.

8. STROKE.—Graduated scales were applied to the frame of the engine above the crossheads at the water end where the stroke indicator is attached, one set on each side, and observations taken of the amount which the engine fell short of making a full stroke at each end, every 2½ minutes throughout the trial. These observations were taken one after the other in rotation, the four readings being obtained by one observer. The variation between different readings was usually confined to less than .001 of a foot. It may be added here that when the engine had once been adjusted, there was scarcely any change made in the dash relief valves, or other mechanism, throughout the whole progress of the main trial. Observations were taken every fifteen minutes of the "stroke register" attached to each side ot the engine, this apparatus being the same as that used by Professor Denton on the Standard Oil Company's engine, and described in the Transactions, A. S. M. E., Vol. XII. The two instruments showed quite different indications, although the length of the strokes of the two sides, as shown on the graduated scales, were practically identical. Neither instrument was calibrated, and, consequently, the record of the readings has been omitted.

9. INDICATOR DIAGRAMS.—A set of indicator diagrams was taken from each end of each cylinder, both steam and water, at intervals of about half an hour during the test. They were taken one after the other in rotation by one person. (See page 194). One indicator was used for all four ends of the high-pressure cylinders, being transferred from one point to the other, as required, and another instrument was used for all four ends of the water cylinders. A third indicator was used for one low-pressure cylinder, and a fourth for the other cylinder.

THE WEIR TEST.

The weir is the same as that which was erected at the time when the water-works were first built. The crest of the weir consists of a wrought-iron plate about one-fourth of an inch thick, with a square edge and vertical ends, attached to a wooden backing with bevelled edges. Its average length is 4.279 feet. The chambers in front end beyond the weir have a depth of about 22 feet below the crest. The width of the chamber in front of the weir is 8 feet. At a distance of 4 feet 6 inches back, the width is reduced to 5 feet 6 inches, the reduction occurring wholly on one side. The hook-gauge box, which, in the original experiments was located in front of the gate-house, was, for the purposes of these tests, placed beyond the weir, at a distance far enough to clear the stream of water falling over it. The pipe leading to it was of the ¾-inch size, and it was laid 10 inches below the crest of the weir, extending back from its face a distance of 2 feet 10 inches. The pipe in front of the weir was 7 feet long, and this was perforated with 40 ⅛-inch holes. For the purposes of these tests a vertical screen was placed in front of the weir at a distance of 3 feet 5 inches, and a horizontal screen from the bottom of the same to the face of the weir, at a depth of 4 feet 2 inches below its crest. The screen was made of wooden strips ½ inch thick and 5 inches wide, laid side by side ½ inch apart, the direction being parallel to the current of water The general form of the chambers, and the location of the various parts referred to, are shown in the accompanying sketches.

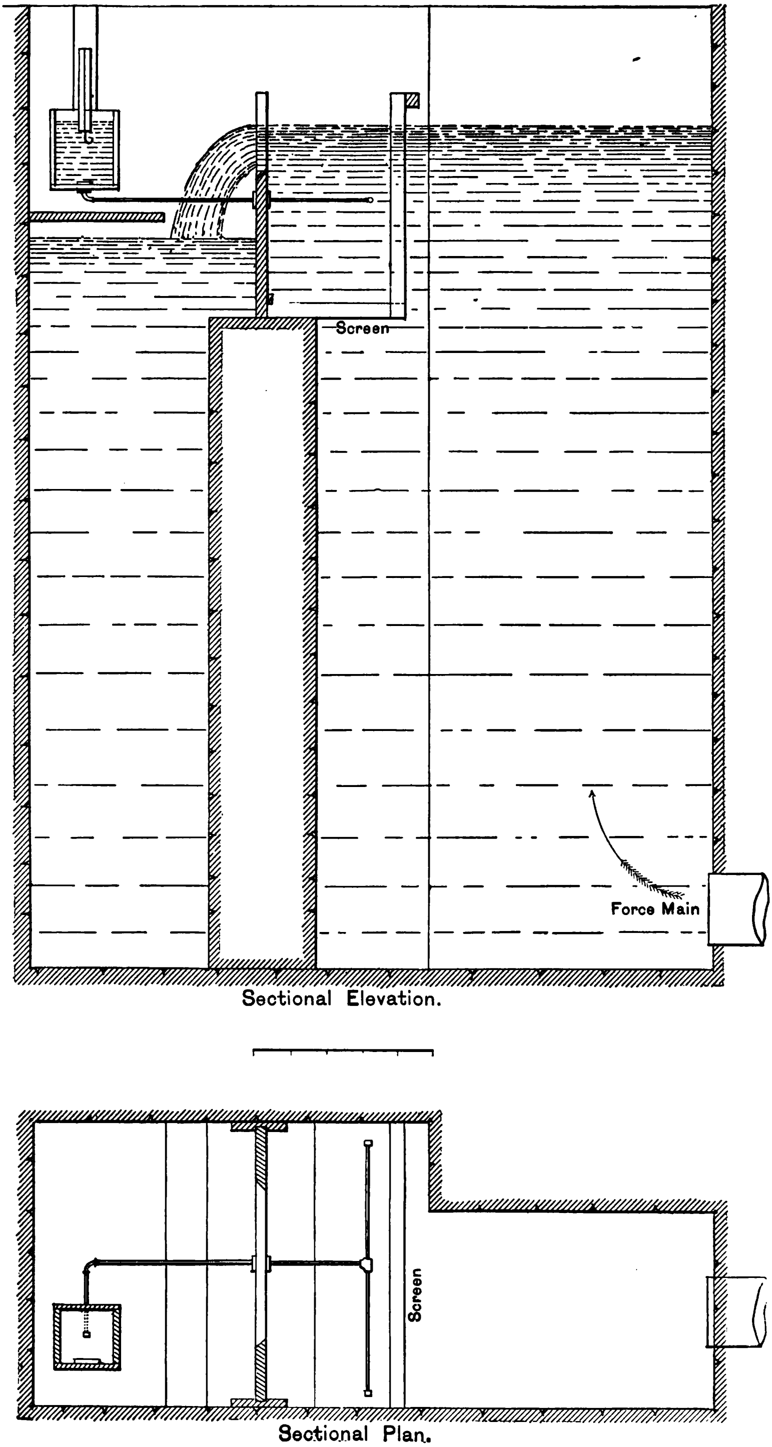
In preparation for the weir test, the 12-inch gate in the pipe leading from the force main to the distributing service of the city was shut, and its condition as to tightness was determined by drawing off the water from the force main, and observing what leaked through at the blow-off valve. It was found to be practically tight. When the water was thus drawn off, the sluice gate between the outlet and inlet chambers at the reservoir was found to leak. As it was necessary in taking the readings of the hook-gauge to keep the water in the outlet chamber some 2 feet below the crest of the weir, the leakage of the sluice gate furnished an opportunity for some of the water discharged from the engine to find its way into the reservoir without passing over the weir. The tightness of the check valve in the force main enabled the extent of this leakage to be determined by observing the rate at which the water fell away in the inlet chamber after the engine had ceased pumping. It was found that this leakage, compared with the total quantity pumped, was insignificant, being but a very small fraction of 1 per cent. As to the tightness of the gate in the main leading from the Morris Engine, and of the check valve in that leading from the old Worthington Engine, it was found in the case of the former that, after the water was drawn off from the pipe between the gate and the engine, there was a stream of water leaking through, without pressure, having a size less than ½-inch in diameter, and, in the case of the latter, absolutely no water appeared in the open force main chamber of the engine.

During the progress of the test, observations of the hook-gauge at the weir were taken every 5 minutes. At the time of each observation, the reading taken was the average of four observations made at intervals of one-quarter of a minute. During the 10-hour period of the main duty trial, the readings varied from a minimum of 1.321 to a maximum of 1.351, and the average was 1.3421. The zero point of the weir was at a height of .186, shown by the hook-gauge, thus leaving 1.1561 feet as the average head on the weir for the 10-hour test. The zero reading was carefully obtained by the use of a surveyor's level. The formula used in computing the discharge is the Francis formula, $Q = 3.326\ (L - O.2\ H.) \times H^{\frac{3}{2}}$, in which Q is the number of cubic feet discharged per second, L, length of the weir, and H, the head on the weir. This is corrected for the velocity of approach, in the usual manner, the area used for determining this velocity being 67.2 square feet. The quantity of water discharged in 24 hours, thus computed, amounts to 10,867,392 gallons. Applying the correction for leakage of the sluice gate, which amounted to 8,616 gallons, the total volume of water, according to the measurements, amounted to 10,876,008 gallons in 24 hours.

Owing to the narrow width of the approaching channel, the short length of the weir for the quantity of water used, and the effect of the projecting corner of the wall on one side, in front of the weir, and the short distance of the hook-gauge pipe from the face of the weir, all of which were unavoidable in this case, the weir test must be considered as a less accurate measurement of the actual quantity discharged than it would have been if these objectionable features had not existed.

PERSONS ENGAGED IN THE TESTS.

The tests were under the direction of the writer, representing the Water Board, and they were made in the presence of Mr. J. J. DeKinder, M. E., of Philadelphia, who represented the builders of the engine. The measurements of feed water and the indicator



SKETCH SHOWING GENERAL FEATURES OF INLET CHAMBER AND WEIR, AS ARRANGED FOR THE TEST. SCALE $\frac{3}{16}$ IN. 1 FT.

diagrams were taken by assistants of the writer. All the other observations were taken by Mr. George Bowers, City Engineer, aided by members of the staff of assistants from his office.

CAPACITY TEST.

On the conclusion of the duty trial proper, between the hours of 8 and 9 P. M., the cut-off valves of the engine were adjusted, so as to increase its speed, and determine whether the engine is capable of fulfilling the terms of the contract relating to capacity. This test was made of short duration, owing to the excessive water pressure developed under the increased load. The speed attained was 24 revolutions per minute, and the engine under these conditions performed its work satisfactorily, there being no evidence of injury to the working parts. The average length of the stroke was 3 per cent. less than that of the main trial, and the quantity of water delivered, measured by plunger displacement, was at the rate of 12,706,583 gallons in 24 hours, which is nearly 6 per cent. in excess of the capacity guaranteed.

MEMORANDA.

All steam and water connections between the new plant and the old plant were either cut off altogether, or blanked off, or it was satisfactorily proved that there was no leakage, either of water or steam, from one plant to the other. The same is true in regard to steam which might have otherwise been wasted by connection with the heating apparatus of the pumping station, and of water which might have been wasted at the blow-off.

All thermometers, gauges, indicator springs, and weighing scales were verified by comparison with standards unless known to be correct.

The valves and pistons of the steam cylinders were subjected to leakage tests with the engine at rest. The valves and pistons of the low-pressure cylinders were found to be practically tight. There was some leakage of the high-pressure pistons. The valves of the high-pressure cylinders, so far as they could be tested, were found in fairly good condition with the exception of the cut-off valve of the No. 1 cylinder (the one farthest from the Morris Engine), nearest the water end. The leakage of the plunger of the No. 2 cylinder (the one nearest the Morris Engine), when subjected to 50 pounds water pressure, was observed, though not measured. The leakage at the point in the stroke where it was greatest (that is, at the end) was easily carried off by the ¾-inch drain valve, and its quantity was, therefore, too small to produce any material amount of slip.

RESULTS OF THE TESTS.

The results of the tests are presented in the following charts, diagrams and tables:

Chart No. 1 shows graphically the coal and water measurements, and the readings of the counter for the whole trial. The coal measurements refer to the quantities which had been fired at the times indicated. These do not include the weight of wood used in lighting the fires. The water measurements show the quantity of main feed water which had been supplied for every twenty emptyings of the weighing tank, uncorrected for differences in the level of the water in the boilers. The measurements for the jacket water are the quantities which had been drawn off at intervals of two hours. The revolutions given (or the number of quadruple displacements) present the hourly record.

Chart No. 2 gives the observations of the boiler gauge, the gauges on the force main and suction main (those of the writer), the temperature of the main feed water, the temperature of the escaping gases so far as authentic, and the indications of the draught gauge.

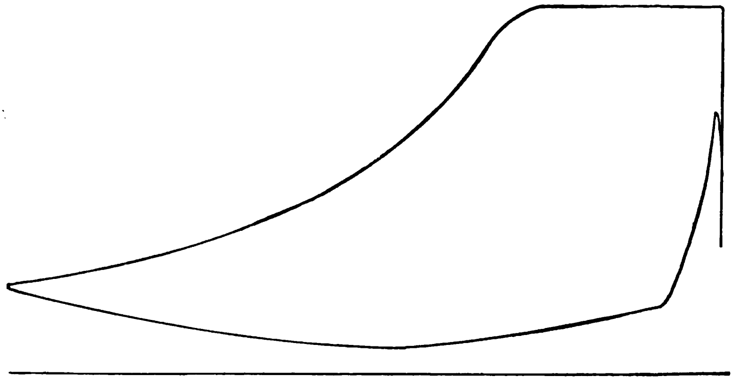
The page of diagrams given are samples of those taken during the 10-hour duty run. There is so great a similarity between the diagrams taken from the different ends of the different cylinders, that only one representative set is here represented.

Table No. 3 gives the data, averages of observations, and results of the engine test, or duty trial proper.

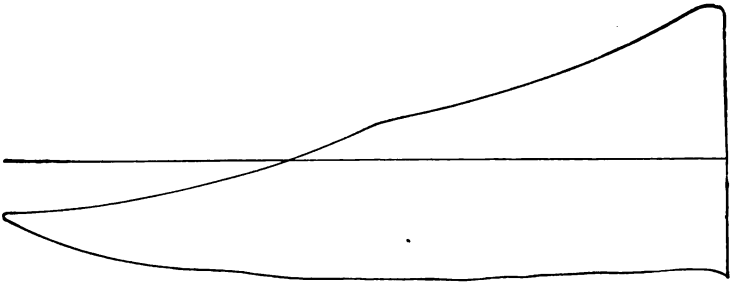
Table No. 4 gives the data, averages of observations, and results pertaining to the work of the boilers.

REPRESENTATIVE INDICATOR DIAGRAMS, FEB. 12TH, 1892.

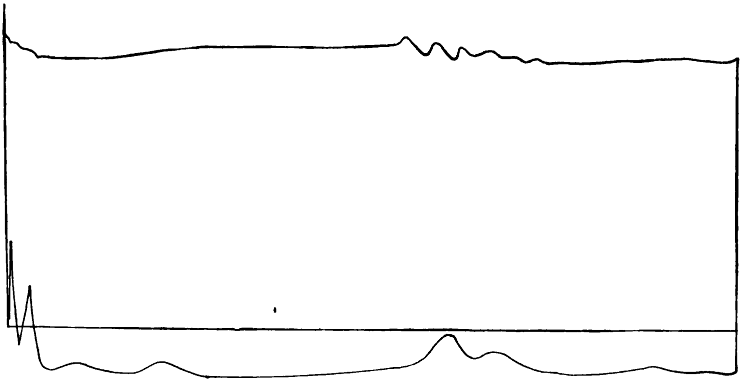
Time	3 P. M.	Vacuum gauge	25.9 ins.
Pressure in steam pipe . .	96 lbs.	Gauge on force main . .	77 lbs.



No. 2 H. P. Cylinder—end farthest from the pump. Scale 48.



No. 2 L. P. Cylinder—end nearest pump. Scale 20.



No. 2 Pump Cylinder—end farthest from steam cylinders. Scale 50.

TABLE NO. 3.—DUTY TRIAL OF ENGINE.

DIMENSIONS.

1.	Net area of each plunger,	578.49 sq. ins.
2.	Net area of each high-pressure steam piston,	470.23 “
3.	Net area of each low-pressure steam piston,	1,953.7 “
4.	Average length of stroke of all pistons during the trial,	3.142 ft.

TEMPERATURES.

5.	Temperature of water in pump well,	34 degs.
6.	Temperature of water supplied to the boilers by the main feed pump, as determined by the preliminary test,	170 “
7.	Temperature of water supplied to the boilers by the jacket pump,	326 “

FEED WATER.

8.	Weight of water supplied to the boilers by the main feed pump,	57,046 lbs.
9.	Weight of water supplied to the boilers by the jacket pump,	8,084 “
10.	Weight of water drained from the separator,	84 “
11.	Total weight of feed water supplied to the boilers from all sources,	65,214 “

PRESSURES.

12.	Boiler pressure indicated by gauge attached to steam pipe (corrected),	96.2 lbs.
13.	Pressure indicated by gauge on force main,	77.33 “
14.	Vacuum indicated by gauge on suction main,	12.32 ins.
15.	Pressure corresponding to vacuum given in preceding line,	6.05 lbs.
16.	Pressure equivalent to distance between the two gauges,	1.68 “
17.	Total water pressure,	85.06 “

MISCELLANEOUS DATA.

18.	Duration of trial,	10 hrs.
19.	Total number of revolutions during trial,	12,516
20.	Total number of single strokes during trial,	50,064
21.	Percentage of moisture in the steam supplied to the engine,	0.38 per ct.

22.	Mean effective pressure measured from high-pressure steam diagrams,	43.14 lbs.
23.	Mean effective pressure measured from low-pressure steam diagrams,	15.53 lbs.

PRINCIPAL RESULTS.

24.	Duty based on 1,000,000 heat units,	116,208,007 ft.-lbs.
25.	Capacity in 24 hours by plunger displacement,	11,345,141 gals.
26.	Capacity in 24 hours by weir measurement,	10,876,008 “
27.	Percentage of slip,	4.1 per ct.
28.	Percentage of total friction,	2.8 “

ADDITIONAL RESULTS.

29.	Number of revolutions of engine per minute,	20.86
30.	Indicated horse-power developed by the high-pressure steam cylinders,	161.16
31.	Indicated horse-power developed by the low-pressure steam cylinders,	241.01
32.	Indicated horse-power developed by all the steam cylinders,	402.17
33.	Feed water consumed by the plant per hour,	6,521.4 lbs.
34.	Feed water consumed by the plant per I. H. P. per hour corrected for moisture in the steam,	16.13 lbs.
35.	Number of heat units consumed per I. H. P. per hour,	16,561.8 B. T. U.
36.	Number of heat units consumed per I. H. P. per minute,	276 B. T. U.
37.	Steam accounted for by indicators at cut-off H. P. cylinder,	9.96 lbs.
38.	Steam accounted for by indicators at release H. P. cylinder,	11.56 “
39.	Steam accounted for by indicators at cut-off L. P. cylinder,	13.01 “
40.	Steam accounted for by indicators at release L. P. cylinder,	13.05 “
41.	Proportion which steam accounted for by the indicators bears to the feed water consumption, at cut-off H. P. cylinder,	.617
42.	Proportion which steam accounted for by the indicators bears to the feed water consumption, at release H. P. cylinder,	.717

43.	Proportion which steam accounted for by the indicators bears to the feed water consumption, at cut-off L. P. cylinder,	.806
44.	Proportion which steam accounted for by the indicators bears to the feed water consumption, at release L. P. cylinder,	.809
45.	Proportion which the jacket water bears to total feed water consumption,	.122
46.	Work done by 100 lbs. of coal, computed for the whole time which the engine was in operation, after correcting for the heat lost by throwing away the jacket water,	110,437,241 ft.-lbs.

NOTE: The weights of feed water used by the engine are corrected for the steam consumed by the calorimeter.

TABLE NO. 4.—BOILER TEST.

47.	Duration of trial,	13.79 hrs.

AVERAGE PRESSURES AND TEMPERATURES.

48.	Steam pressure by engine-room gauge (corrected),	96 lbs.
49.	Atmospheric pressure by barometer,	14.7 "
50.	Force of draft in inches of water,	.14 ins.
51.	Temperature of escaping gases,	486 deg.
52.	Temperature of feed water,	106 "

FUEL.

53.	Weight of wood used in lighting fires,	378 lbs.
54.	Equivalent value of wood expressed in coal,	151.2 "
55.	Total weight of dry coal consumed, including wood equivalent,	9,718.4 "
56.	Weight of ashes,	789.2 "
57.	Weight of combustible consumed,	8,929.2 lbs
58.	Percentage of ashes,	8.1 per cent.
59.	Coal consumed per hour,	704.8 lbs.
60.	Coal consumed per hour per sq. ft. of grate,	9.79 lbs.
61.	Total heat of combustion of the coal obtained by calorimeter trial,	13,361 B. T. U.

QUALITY OF STEAM.

62.	Percentage of moisture in steam at engine, including separator water,	.51 per cent.
63.	Quality of steam generated at boiler,	assumed dry

WATER.

64.	Total weight of water evaporated,	90,479 lbs.
65.	Weight of water evaporated per hour,	6,560.4 “
66.	Equivalent water evaporated per hour from and at 212 degrees,	7,537.9 “
67.	Equivalent water evaporated per sq. ft. of heating surface per hour,	2.1 “

ECONOMIC EVAPORATION.

68.	Water actually evaporated per pound of dry coal from actual pressure and temperature,	9.31 lbs.
69.	Water actually evaporated per pound of dry coal from actual pressure and temperature, corrected for heat lost by throwing away jacket water,	9.549 “
70.	Equivalent water evaporated per pound of dry coal from and at 212 degrees,	10.697 “
71.	Equivalent water evaporated per pound of combustible from and at 212 degrees,	11.64 “
72.	Number of pounds of coal required to supply 1,000,000 B. T. U.,	96.77 “
73.	Coal per I. H. P. per hour (lines 34 and 69),	1.69 “

CONCLUSIONS.

1. CAPACITY.—The results of the tests show that the requirements of the contract in the matter of capacity are completely fulfilled. Throughout the duty trial, the average capacity was at the rate of 11,345,141 gallons in 24 hours, which is 13.4 per cent. in excess of the nominal capacity of the engine; and during the special run elsewhere referred to, the capacity was increased to a point 27 per cent. in excess of the nominal rate, and 7 per cent. more than the 20 per cent. increase above the nominal capacity, as required by the contract. In considering these figures, it must not be forgotten that the engine performed its work against a greater resistance than that originally intended, the water being discharged through a 24-inch main, whereas the capacity stipulated in the contract was based upon the use of a 30-inch main. The use of the existing force main increases the pressure over that which would occur if a 30-inch main had been used about 5 pounds.

2. SLIP.—According to the figures shown by the weir test, the slip of the pumps amounted to 4.1 per cent. For the reasons given in the body of the report, under the head of “The Weir Test,” it is believed that the weir measurement did not show the total quantity

of water actually discharged to the reservoir, and the slip, in reality, was less than the apparent quantity found. There are other and stronger reasons for this belief. In the old Worthington engine, which I tested, and which is reported in another communication, the slip, when pumping at the rate of about 6,000,000 gallons in 24 hours (for the measurement of which quantity the weir is more reliably adapted), was found to be about 3 per cent. It is unreasonable to suppose that the new pump would show a larger amount of slip than the old one which has seen many years of service, especially when it is considered that the leakage of one of the plungers of the old pump, which was tried, was much in excess of that shown on a similar trial of the new engine. Taking the percentage of slip as found, however, it comes practically within the limit imposed by the contract, which was 4 per cent., and the requirements of the agreement in this respect may therefore be considered fulfilled.

3. DUTY.—The duty performed by the engine, as computed from the data, is 116,208,007 foot-pounds, referred to heat units, and this is well above the quantity guaranteed in the contract. It may be added that the work done referred to the old basis of "100 pounds of coal" is also above the quantity required, considering the total amount of coal burned, and the work done during the whole time that the engine was in operation, in all nearly 14 hours.

4. BOILER PERFORMANCE.—The economy shown by the work of the boilers—*viz.*, 11.64 pounds of water evaporated from and at 212 degrees per pound of combustible, is creditable, though the result is not so high as that often obtained from boilers of similar type. An evaporation of 12 pounds is considered a standard for good bituminous coal, and it sometimes reaches 12.5 pounds. The quality of the coal, as indicated by the heat test, is somewhat inferior to that of the best class of bituminous coals. The best coals give between 13,500 and 14,000 B. T. U. on the heat test. The coal used on the duty trial (which was probably a mixture of New River and Cumberland) gave 13,361 B. T. U.

5. IN GENERAL.—The new pumping engine conforms in all respects to the requirements of the contract, and it is in every way an admirable and satisfactory machine. The whole plant, embracing boilers, engine and accessories, furnishes the desired increase in capacity to the Water-Works by means which are creditable alike to the Water Board and to the City of Lowell.

I am, faithfully yours,

GEO. H. BARRUS.

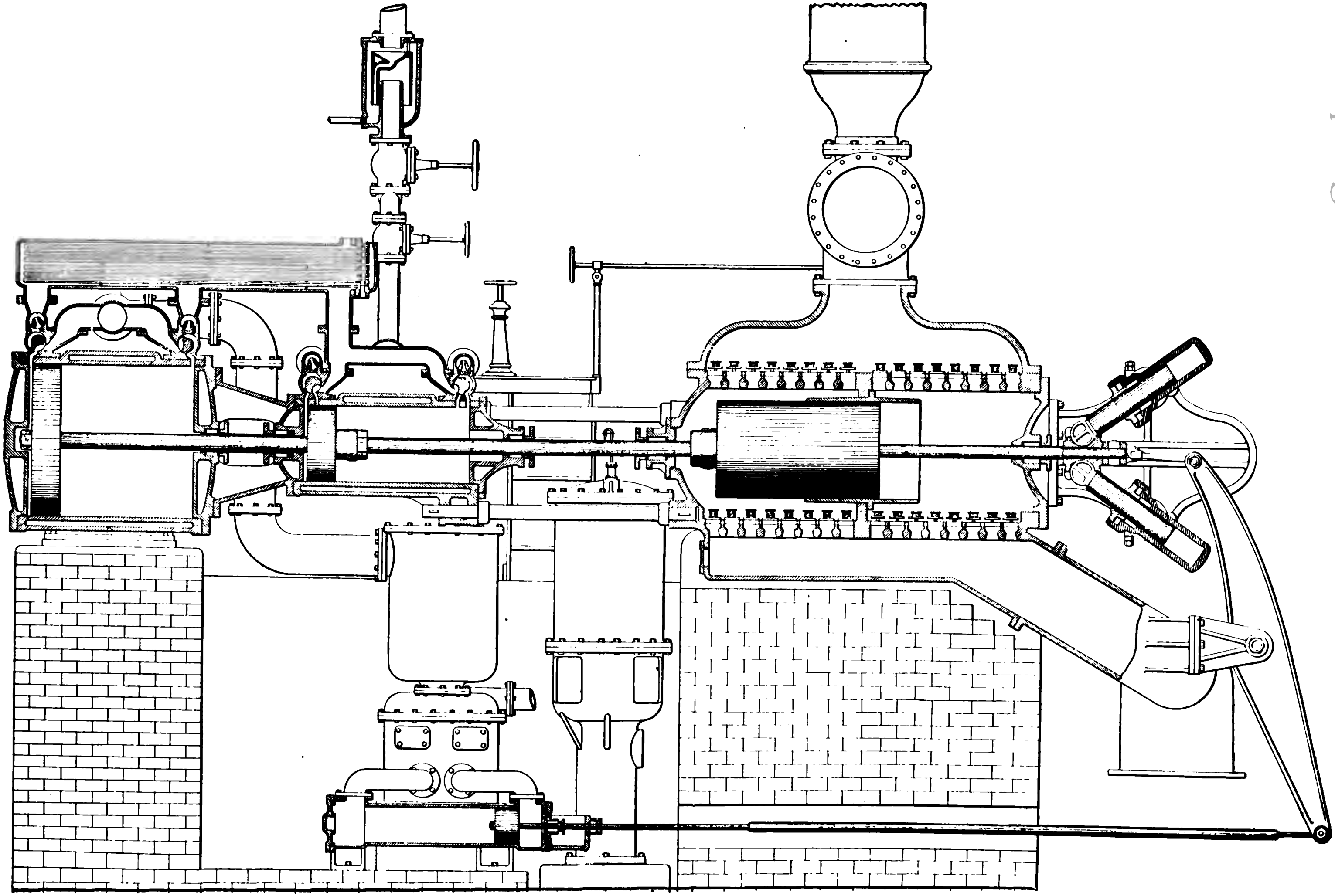


CHART No. 1.—COAL, WATER, AND REVOLUTIONS.

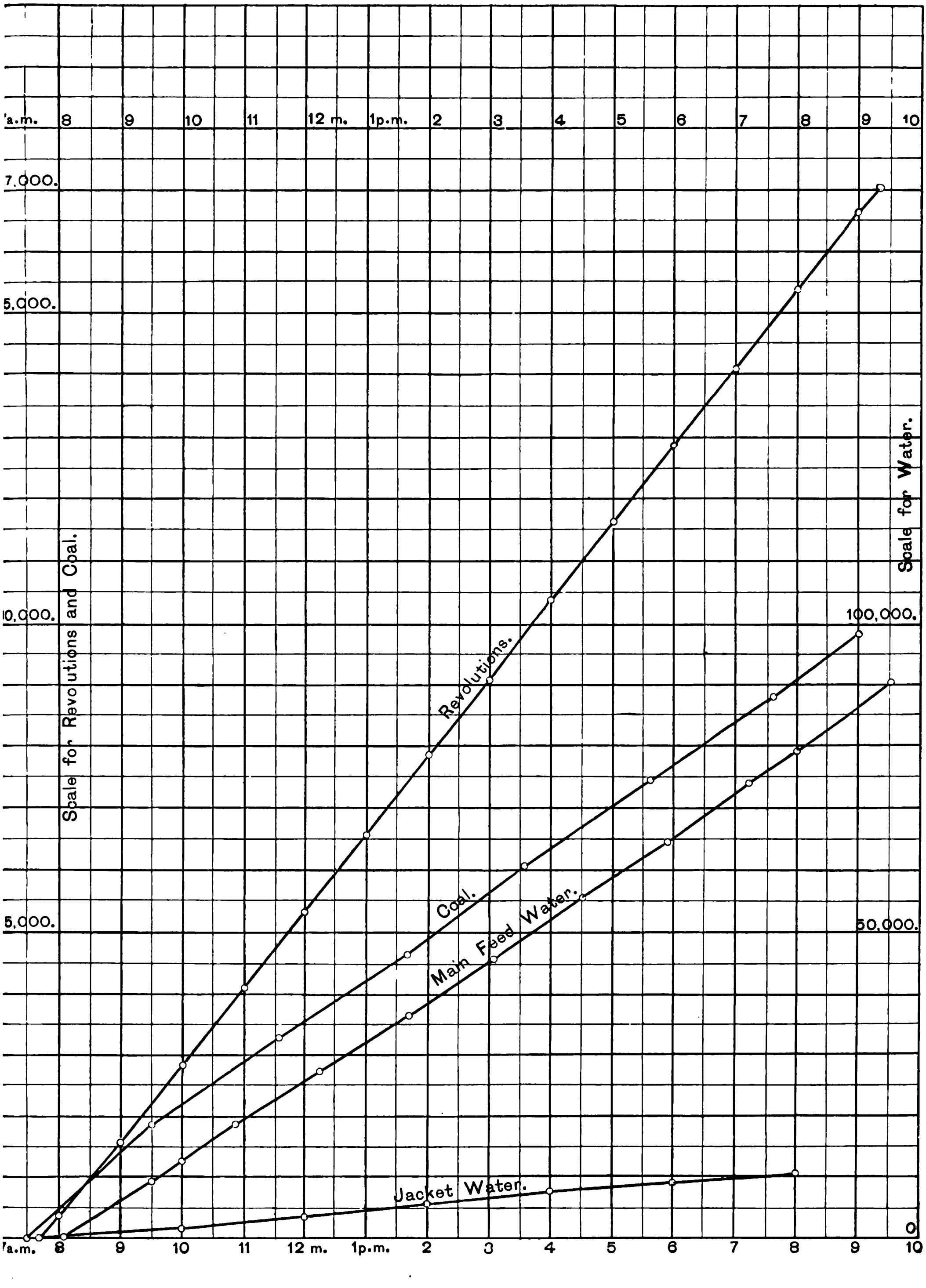
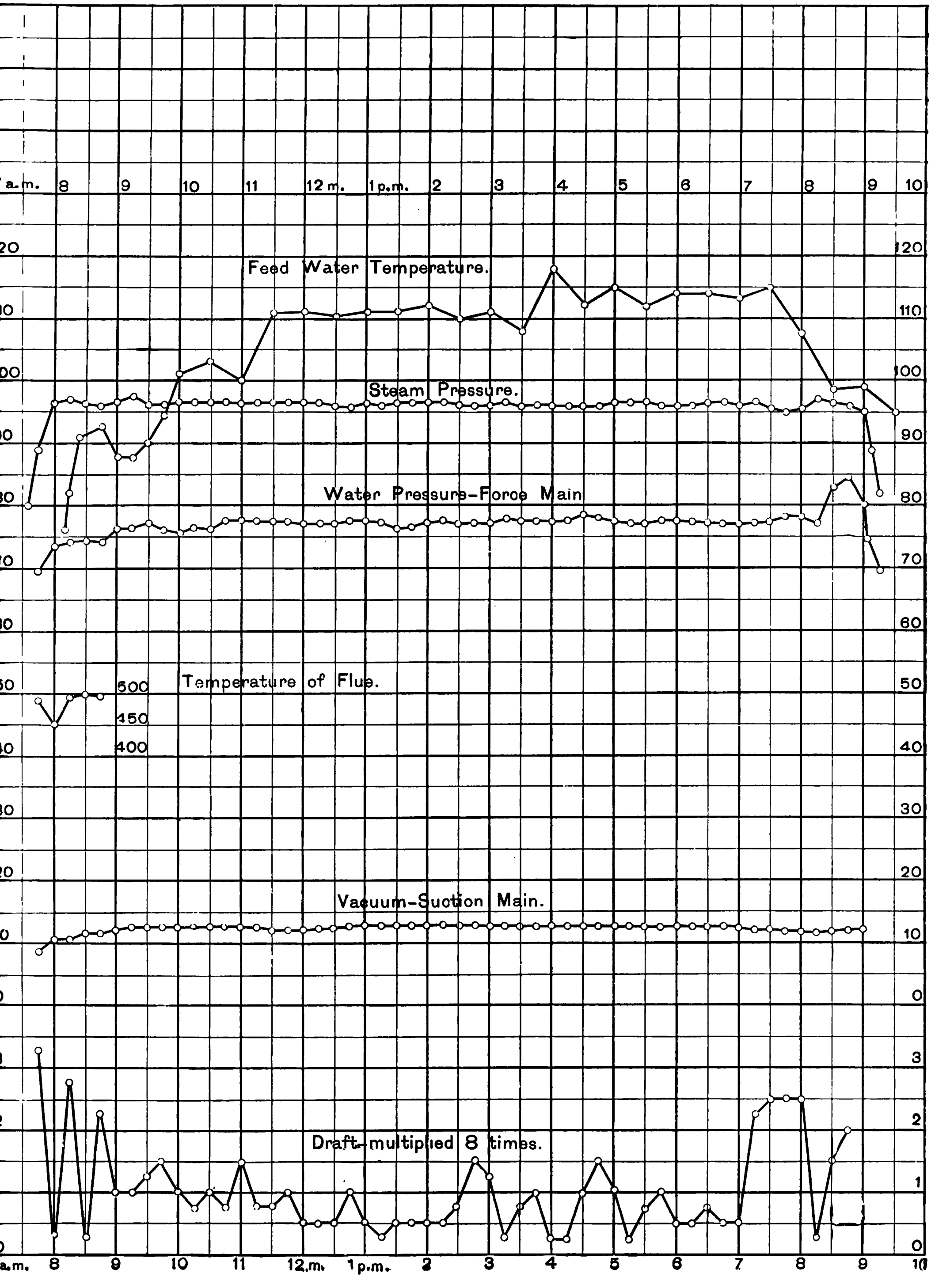


CHART No. 2.—PRESSURES, TEMPERATURES, AND DRAFT.



PORT PERRY, PA.

3,000,000 GALLONS CAPACITY.

ONE ENGINE.

REPORT OF JACOB SCHINNELLER, CIVIL ENGINEER.

To J. R. McGinley, *President Turtle Creek Valley Water Co.:*

Dear Sir—I herewith submit report of a test of the triple-expansion pumping engine erected by Henry R. Worthington for the Turtle Creek Valley Water Company, at their pumping station in Port Perry, Pa. The engine is of the Worthington high-duty triple-expansion vertical type, and is fitted with compensating cylinders, surface condenser and automatic jacket drain attachment.

It is the first high-duty triple-expansion duplex pumping engine ever constructed.

CONTRACT REQUIREMENTS.

The contract requires that the engine shall be capable of pumping at the rate of three million (3,000,000) U. S. gallons of water per twenty-four (24) hours, against two hundred and ninety (290) feet head, including suction lift, at a piston speed not to exceed one hundred (100) feet per minute, and develop a duty of one hundred and ten million (110,000,000) foot-pounds per one thousand (1,000) pounds of steam consumed.

CONDITIONS OF TEST.

Steam was furnished by the new Root boiler, burning natural gas, and the feed-water was measured by a brass Worthington meter, which was carefully tested.

The engine was run under regular conditions of service, pumping into the reservoir.

The engine ran very smooth and uniform, and required very little care from the attendant.

ENGINE DATA.

High-pressure steam cylinders, diameter,	15 ins.
High-pressure steam cylinders, number,	2
Intermediate-pressure cylinders, diameter,	33 ins.
Intermediate-pressure cylinders, number,	1
Low-pressure steam cylinders, diameter,	57½ ins.
Low-pressure steam cylinders, number,	1
Double-acting water plungers, diameter,	16½ ins.
Double-acting water plungers, number,	2
Stroke of piston and plunger, nominal,	36 ins.
Stroke of piston and plunger, actual,	37.82 ins.

RESULTS OF TEST.

Date of test,	March 9th, 1892.
Duration,	8 hours.
Total number of strokes of plungers,	30,416
Average length of stroke,	37.66 ins.
Displacement of plunger per stroke,	33.96
Piston speed in feet per minute,	99.41
Rate of pumping in 24 hours,	3,100,000 gals.
Head observed by water gauge,	112.0 lbs.
Head observed by suction gauge,	4.94 “
Head due difference in level of gauges,	9.33 “
Total head,	126.27 “
Average steam pressure,	139.85 “
Average vacuum,	26.45 ins.
Total number of expansions of steam,	21.70 “
Temperature of main well,	40.5°
Temperature of feed water,	176.5°
Temperature of air-pump delivery,	51.0°
Temperature of jacket water,	355.5°
Temperature of engine room,	72.0°
Temperature of outside air,	50.0°
Total quantity of feed registered by meter,	18,762.0 lbs.
Total quantity of feed returned from jackets,	2,899.1 “
Equivalent feed-water reduced to temperature of well,	16,598.0 “
Equivalent jacket water reduced to temperature of well,	2,118.0 “
Total feed to be charged to boilers,	18,716.0 “
Net feed to be charged to engines,	18,719.0 “

Dryness of steam, - - - - - - - 100 per ct.

Weight of a gallon of water at 40.5°, - - 8.3455 lbs.

$$\text{Duty} = \frac{\overset{\text{(Total ft.-lbs.)}}{2{,}510{,}377{,}080} \times 1{,}000}{18{,}719} = \text{ - - - - } 134{,}025{,}993 \text{ ft.-lbs.}$$

It will be seen from the above results that the requirements of the specifications as to capacity has been complied with, and the duty actually obtained has exceeded duty guaranteed by 24,025,993 foot-pounds, or 21.84 per cent.

At above rate of speed and duty, there would be used under the boiler, 22,462 cubic feet of natural gas for each one million (1,000,000) gallons of water pumped into the reservoir, allowing 3 per cent. for imperfect filling of pumps from slip, leakage past the plunger, etc.

As the pumps are new, and valves in good condition, the above 3 per cent. will more than cover any loss from above causes. The amount of water that will leak past the plunger is much less when engine is pumping than when it is at rest. The plunger in motion tends to keep the water back, as it travels in the opposite direction from the water trying to pass by it.

Respectfully submitted,

(Signed) J. SCHINNELLER.

NORFOLK, VA.

15,000,000 GALLONS CAPACITY.

TWO ENGINES.

REPORT OF GEORGE W. BAIRD, U. S. N., ON THE DUTY AND CAPACITY TESTS OF THE HIGH SERVICE PUMPING ENGINES, OF THE CITY OF NORFOLK, VA., THE 10TH AND 11TH MAY, 1892.

The Board of Water Commissioners, Norfolk, Va.

GENTLEMEN—I have the honor to report that the new high service pumping engines have been tested in accordance with the terms of the contract, and that the following is a description of the plant and a record of the test. In conducting this test, I have had the assistance and hearty co-operation of the City Engineer and his assistants, and of the Chief Engineer of the Water Works, and Mr. K. McAlpine, of the Engineer Corps of the Navy.

DESCRIPTION OF THE PLANT.

Boilers.

The boilers are of the horizontal tubular type, set in brickwork and usually denominated "shell boilers." They are externally fired, the gases of combustion pass under the shells and return through tubes, and debouch into a rectangular uptake, and are delivered to a brick chimney.

Number of boilers,	4
Diameter of shells,	66 ins.
Length of shells,	18 ft.
Number of tubes in each boiler,	58
External diameter of tubes,	4 ins.
Length of grates,	5 ft. 6½ "
Width of grates—each boiler,	5 ft. 9 "
Area of grate surface in one boiler,	31.86 sq. ft.
Heating surface in one boiler,	1,263.61 " "
Area through tubes for draught,	4.4251 " "

Ratio of heating surface to grate surface, - -	39.6 : 1
Ratio of grate surface to area through tubes for draught, - - - - - - -	7.2 : 1

The Engines.

There are two (2) compound, duplex, direct-acting, horizontal, pumping engines of the Worthington patent.

One engine is of ten million (10,000,000) gallons capacity, and the other is of five million (5,000,000) gallons capacity, per twenty-four (24) hours. Each engine has two (2) high-pressure cylinders and two (2) low-pressure cylinders and two (2) water cylinders.

Each water cylinder is worked by one high and one low-pressure steam cylinder, the piston rod of which extends through the water end and actuates two opposite pairs of oscillating compression cylinders known as " high-duty attachments," the purpose of which is to absorb part of the power of the steam cylinders during the admission of steam, and to give it out during the period of expansion. These oscillating attachments are filled with water, and as their plungers work in and out, the water regurgitates through their ports and trunnions to the lower end of a large vertical cylinder, the upper portion of which is filled with air, which forms an air cushion.

To replenish the loss of air due to leaks and the absorption of air in the air chambers, a pair of small duplex air compressors are provided ; these, however, were not operated all the time. All the steam cylinders are steam jacketed ; they drain through Nason steam traps into a feed tank, whence the independent feed pump receives its water and delivers it to the boilers.

Each engine has a jet condenser and two horizontal air pumps; the latter are worked by levers from the main engines. The pumps receive their water through mains from the impounding reservoir, and deliver into the city mains against a pressure, ordinarily, of about 40 pounds.

The plungers are of cast-iron covered with brass, working through composition rings without packing. The pump valves are of rubber, working on composition seats and stems pressed down by springs. The usual counters and gauges are provided. The strokes of the engine are nominal ; by regulating the point of cut-off or the amount of steam cushion, or by varying the load on the engines, the strokes of the piston are varied.

The following records the principal dimensions of the engines:

THE SMALL ENGINE*

Capacity of engine per day,	5,000,000 gals.
Number of high-pressure cylinders,	2
Diameter of high-pressure cylinders,	18 ins.
Stroke, nominal,	36 "
Stroke, extreme,	38 "
Mean stroke during test,	37.7203 "
Diameter of high-pressure piston rods,	3½ and 3¾ "
Net area of high-pressure piston,	244.14 sq. "
Displacement of H. P. piston per stroke,	9,209.04 cu. "
H. P. steam passage clearance,	125 " "
Percentage of clearance to piston displacement,	1.35 per ct.
Area through steam port of H. P. cylinder.	11 sq. ins.
Mean clearance of H. P. piston during test,	0.1398 "
Number of L. P. cylinders,	2
Diameter of L. P. cylinders,	36 ins.
Diameter of L. P. piston rod,	3½ "
Net area of L. P. piston,	1,013.07 sq. "
Stroke of L. P. piston (extreme)	38 "
Mean stroke of L. P. piston during test,	37.7203 "
Displacement per stroke of L. P. piston,	38,213.31 cu. ins.
Clearance in passages in one end of L. P. cylinders,	350 " "
Percentage clearance to piston displacement,	0.9 per ct.
Area of low-pressure steam port,	23 sq. ins.
Area of low-pressure exhaust port,	31⅝ " "
Mean clearance of L. P. piston,	0.1398 "
Number of air-pump cylinders,	2
Diameter of air-pump cylinders,	7 ins.
Net area of suction or delivery valves at each end of one air-pump cylinder,	26.85 sq. "
Stroke of air pump, extreme,	38 "
Mean stroke during test,	37.7203 "
Aggregate piston displacement of air pump per revolution of engine.	5,806.46 cu. "
Number of feed pumps,	1 duplex.
Diameter of steam cylinders of feed pump,	6 ins.
Diameter of water cylinders of feed pump,	4 "

Number of main pumps,	2
Diameter of main plungers,	19¼ ins.
Stroke, extreme,	38 "
Mean stroke during test	37.7203 "
Diameter of plunger rods,	4⅜ and 3¼ "
Net area of plungers,	279.36 sq. "
Net displacement of plungers per stroke in gallons,	45.617
Aggregate displacement of plungers per revolution of the pump in gallons,	182.468
Number of suction valves in each end of each water cylinder,	18
Aggregate area of suction valves for each end of each water cylinder,	158.4 sq. ins.
Percentage valve area of plunger area,	55.4 per ct.
Number of delivery valves in each end of each water cylinder,	18
Aggregate area of delivery valves in each end of each water cylinder,	158.4 sq. ins.
Percentage valve area of plunger area,	55.4 per ct.
Number of air chambers,	1
Height of air chamber,	6 ft. 6½ ins.
Diameter of air chamber,	36 "
Number of high-duty attachments on each engine,	2
Diameter of H. D. attachment plungers	6 ins.
Stroke of H. D. attachment plungers,	10¼ "
Number of compression cylinders for each H. D. attachment,	2
Number of duplex air compressors,	1
Diameter of steam cylinders of air compressor,	4½ ins.
Diameter of air cylinders of air compressor,	3 "
Length of stroke of air compressor,	4

THE LARGE ENGINE.

Capacity of the engine in gallons per day,	10,000,000
Number of H. P. cylinders,	2
Diameter of H. P. cylinders,	25 ins.
Stroke, nominal,	48 "
Stroke, extreme,	50 "

Mean stroke during test, - - - -	49.6675 “
Diameter of H. P. piston rods, - - -	5¾ and 5 “
Net area of H. P. pistons, - - - -	468.07 sq. “
Displacement of each H. P. piston per stroke,	23,247.87 cu. “
Clearance in H. P. steam passages, - - -	350 “ “
Percentage of clearance to piston displacement,	1.38 per ct.
Area of steam port of H. P. cylinder, - -	15 sq. ins.
Area of exhaust port of H. P. cylinder, - -	25 “ “
Mean clearance of H. P. piston, - - -	0.16625 “

Number of L. P. cylinders, - - - -	2
Diameter of L. P. piston rod, - - - - -	5 ins.
Diameter of L. P. cylinders, - - - -	50 “
Net area of each L. P. piston, - - -	1,953.682 sq. “
Stroke, nominal, - - - - - - -	48 “
Stroke, extreme, - - - - - - -	50 “
Mean stroke during test, - - - -	49.6675 “
Displacement per stroke of each L. P. piston,	97,034.5 cu. “
Clearance in steam passages at one end of each L. P. cylinder, - - - - - -	1,120 “ “
Percentage of clearance to displacement, -	1.15 per ct.
Area of L. P. steam port, - - - -	52 sq. ins.
Area of L. P. exhaust port, - - - -	62 “ “
Clearance of L. P. piston, - - - -	0.16625 “

Number of air-pump cylinders, - - -	2
Diameter of air-pump cylinders, - - -	11½ ins.
Net area of suction or delivery valves at one end of each air-pump cylinder, - -	53 sq. “
Stroke of air-pump piston, - - - -	38 “
Aggregate air-pump displacement per revolution of the engine, - - - -	15,786.72 cu. “
Number of feed pumps, - - - - -	1 duplex.
Diameter of steam cylinders, - - - -	6 ins.
Diameter of water cylinders, - - - -	4 “
Stroke of feed-pump piston, - - - -	6 “

Diameter of the main pump plungers, - -	26¾ “
Diameter of the pump rods, - - - -	5¾ and 3⅞ “
Net area of each pump plunger, - - -	543.12 sq. “
Net displacement per stroke in gallons, - -	116.775
Plunger displacement per revolution in gallons,	467.1

Number of suction or delivery valves in each end of each water cylinder, - - -	31
Aggregate area of suction or delivery valves in each end of each water cylinder, - -	272.8 sq ins.
Percentage of valve area of plunger area, -	50.23 per ct.
Number of air chambers, - - - - -	1
Diameter of air chamber, - - - -	36 ins.
Height of air chamber,- - - - -	6 ft. 6½ "
Number of high-duty attachments on each engine, - - - - - - - -	2
Diameter of H. D. attachment plunger, - -	7½ ins.
Stroke of plungers of high-duty attachments, -	13⅜ "
Number of duplex air compressors, - -	1
Diameter of steam cylinders of air compressor,	4½ ins.
Diameter of air cylinders of air compressor, -	3 "
Stroke of pistons of air compressor, - -	4 "

THE TERMS OF THE CONTRACT.

According to the terms of the contract the large engine is to be able to deliver at the rate of ten million (10,000,000) and the small engine at the rate of five million (5,000,000) U. S. gallons per day, when running at a piston speed not exceeding one hundred and twenty-five (125) feet per minute, with an effective steam pressure of not less than one hundred and ten (110) pounds per square inch at the engine, and to deliver those respective quantities against a head not exceeding ninety (90) pounds per square inch, inclusive of suction.

The contract also provides that with the above steam pressure, and operating against a head of 75 pounds, inclusive of suction, they shall develop a duty of one hundred million (100,000,000) foot-pounds for each 100 pounds of best "Pocahontas" coal fed into the furnaces.

"The duty shall be calculated from the actual quantity of water pumped, as measured by the plunger displacement, the total load, including suction on the plungers and the actual amount of coal fed into the furnace during the trials, without any allowance whatever." From this it is clear that the plunger displacement, without correction for slip or loss of action, is to be considered as the volume pumped. By the "best Pocahontas coal" I understand the best merchantable coal from that mine; this I am sure has been purchased by the Board; to satisfy the contractors' agents, however, they were permitted to hand-pick the coal and select the best portions.

The coal consumed during the tests was an excellent quality of Pocahontas coal, and was hand-picked for the tests. The lumps alone were selected. A special analysis by Mr. W. A. S. Taylor, gave the following:

Hygrometric water,	.50
Volatile matter other than water,	19.83
Fixed carbon,	75.63
Sulphur,	.48
Ash,	3.54

MANNER OF MAKING THE TESTS.

The scales for weighing the coal were tested by the Sealer of Weights and Measures and found to be correct. The gauges and thermometers, made by reputable makers, and being new and agreeing closely, were assumed to be correct. The indicators had been tested and corrections applied where errors existed. The indicators were checks on the pressure gauges. Three of these indicators were of the Tabor and seven of the Thompson patent. The coal was weighed by one of the clerks from the office of the Water Commissioners. The pressures in the mains and in the rising mains were indicated by Bourdon gauges, and corrections made for height above the center lines of the pump cylinders. The height of the center line of the pumps, above the surface of the water in the impounding reservoir, from which the water was pumped, was carefully leveled up by the City Engineer and his assistants. The pumps were kept on their regular duty, the excess of output being permitted to escape into the reservoir through a 6-inch pipe.

The terms of the contract made no reference to the potential nor economic vaporization in the boilers, and the contractor's agent would not go to the expense of erecting the usual apparatus for weighing the feed water, but he provided a Worthington water meter for that purpose.

The contractor's engineer ran the engines and the boilers, without interference from myself or the officers of the Water Works. The coal was weighed and delivered to the fire-room in charges of three hundred (300) pounds. The fires were regulated by the dampers and ash-pit doors; these were kept nearly closed, as the rate of combustion was but four and one-half pounds of coal per square foot of grate during the test of the small engine, and 5.96

pounds per square foot of grate during the test of the large engine. The increased pressure in the main was obtained by throttling a valve on the main, near the engine house.

The height of water in the gauge glasses, and the condition of the fires were noted at the beginning of the test and were brought to the same standard at the end of the test.

The steam jackets drained into the feed tanks through a Nason steam trap; as the trap worked intermittently, permitting considerable steam to escape, volumes of hot water and steam were periodically discharged into the feed tank, which occasioned some loss and annoyance. Two boilers were used to test the small engine, and three to test the large engine. The pressures were recorded every 15 minutes. The counters and temperatures were recorded hourly. A set of indicator diagrams were taken hourly.

TEST OF THE SMALL ENGINE.

Date of the test, - - - - - - -	May 10th, 1892.
Duration of test in hours, - - - - - -	10
Mean height from the surface of the water in the reservoir to the center of the pump, -	10.15175 ft.
Height of water-pressure gauge above center of pump, - - - - - - - -	5.73 ft.
Sum of the two preceding quantities, - -	15.88175
Head in pounds per square inch due to the above, $\frac{15.88175}{2.31}$, - - - - - - - -	6.874
Mean head in pounds per square inch, as recorded by the water gauge, - - -	70.737
Aggregate head in pounds per square inch, -	77.611
Mean steam pressure in pounds per square inch above the atmosphere, per engine-room gauge, - - - - - - - -	108.04
Mean vacuum, in inches of mercury, in the condenser, - - - - - - - -	27.26
Mean stroke of the piston in feet, - - -	3.1433
Total number of double strokes (revolutions), -	12,515
Mean number of double strokes (revolutions) per minute, - - - - - - -	20.8583
Number of gallons per double stroke (revolution) displaced by the plungers, - -	182.468

Total number of gallons delivered in the 10 hours based on plunger displacement, (182.468×125.15),	2,283,586
Total number of pounds of coal consumed,	2,865
Total number of pounds of steam consumed per hour, in the low-pressure cylinders, calculated from the pressure there plus 7.4 per ct.	2,929.932
Mean indicated horse-power developed by the engine,	194
Mean temperature of the water in the well main,	65.6
Mean temperature of the injection water,	65.6
Mean temperature of the water discharged from the condenser,	106.63
Mean temperature of the feed water,	155.90
Mean temperature of the external atmosphere,	68.09
Mean temperature of the engine room,	78.91
Total number of pounds of water fed to the boilers, calculated from the terminal pressures in the low-pressure cylinders plus 7.4 per ct.,	29,299.32
Number of thermal units imparted to the feed water per hour, computed,	3,017,829.96
Mean number of pounds of water consumed per indicated horse-power per hour, computed from the steam used in the L. P. cylinder,	15
Mean number of pounds of water vaporized per pound of coal, computed,	10.23
Mean number of pounds of water that would have been vaporized from, and at 212 degrees,	10.906

Duty of the Small Engine.

$$\frac{182.468 \times 8.\tfrac{1}{3} \times 12{,}515 \times 77.611 \times 2.31 \times 100}{2{,}865} = 119{,}081{,}758$$

foot-pounds per one hundred (100) pounds of coal.

The mean speed of the pistons during the test was $(20.8583 \times 3.1438 \times 2) = 131.128$ feet per minute.

At a speed of one hundred and twenty-five (125) feet per minute, of the plungers, the displacement would be,

$$\frac{279.36 \times 2 \times 12 \times 125 \times 1{,}440}{231} = 5{,}224{,}395$$ gallons in twenty-four (24) hours.

Capacity of the Small Engine.

The capacity of the small engine (the volume displaced by its plungers, supposing there is neither slip nor loss of action) according to the terms of the contract was,

$$\frac{182.468 \times 12,515 \times 24}{10} = 5,480,608 \text{ gallons.}$$

TEST OF THE LARGE ENGINE.

Date of Test,	May 11th, 1892.
Duration of the test in hours,	10
Mean height from the surface of the water in the reservoir to the center of the pump in feet,	10.20625
Height of the water pressure gauge above the center of the pump in feet,	6.66
Sum of the two preceding quantities,	16.86625
Head in pounds per square inch due to the above, $\frac{16.86625}{2.31} =$	7.301
Mean head in pounds per square inch recorded by the water gauge,	71.89
Mean aggregate head in pounds per square inch,	79.191
Mean steam pressure in pounds per square inch above the atmosphere per engine-room gauge,	111.22
Mean vacuum in the condenser in inches of mercury,	25.53
Mean stroke of the piston in feet,	4.1389
Total number of double strokes (revolutions) made by the engine,	9,690
Mean number of double strokes (revolutions) of the engine per minute,	16.8766
Number of gallons displaced per double stroke (revolution) of the engine,	467.102
Total number of gallons displaced by the plungers (467.1 × 9,690),	4,526,199
Total number of pounds of coal consumed,	5,700
Mean temperature of the water in the well main,	70.2 deg.
Mean temperature of the injection water,	70.2 “
Mean temperature of the water discharged from the condenser,	100.9 “
Mean temperature of the feed water,	151.18 “
Mean temperature of the external atmosphere,	74.1 “
Mean temperature of the engine room,	79.5 “

Total number of pounds of water fed to the boilers, calculated from the terminal pressures in the low-pressure cylinders, plus 7.4 per ct., - - -	62,865,282
Number of thermal units imparted to the feed water per hour, calculated from the above computed weight of feed water and the range of temperature, - - - - - - - - - -	6,712,440
Mean number of pounds of water consumed per indicated horse-power per hour, computed, - -	14.78
Mean number of pounds of water vaporized per pound of coal, computed, - - - - - - - -	11.03
Mean number of pounds of water, that would have been vaporized per hour, per pounds of fuel had the feed water been delivered at 212 degrees,	12.79
Total weight of steam consumed per hour in the low-pressure cylinders, calculated from the pressure there, plus 7.4 per ct., - - - - - -	6,286.520
Mean indicated horse-power developed by the engine,	425.44
Number of pounds of coal per indicated horse-power, per hour, - - - - - - - - - -	1.34

Duty of the Large Engine.

$$\frac{467.1 \times 8.33 \times 9{,}690 \times 79.191 \times 2.31 \times 100}{5{,}700} = 121{,}050{,}155$$

foot-pounds per hundred (100) pounds of coal, based on plunger displacement with no allowance for slip nor loss of action.

The mean speed of the pistons during the test was (16.8766 × 4.1389 × 2 —) = 139.7 feet per minute. At a mean speed of one hundred and twenty-five (125) feet a minute, the plungers would displace

$$\frac{543.12 \times 2 \times 125 \times 12 \times 1{,}440}{231} = 10{,}157{,}049 \text{ gallons.}$$

Capacity of the Large Engine.

The capacity of the large engine, running at a piston speed employed during the test, the quantity being measured by the plunger displacement without any allowance for slip nor loss of action would be,

$$\frac{467.1 \times 9{,}690 \times 24}{10} = 10{,}862{,}877 \text{ gallons in twenty-four (24) hours.}$$

The surplus of water pumped during the above test was permitted to escape, partly from the fire plugs in the city, and partly from a 6-inch main into the impounding reservoir. During the trial of the small engine (on the 10th inst.) a main on Park Avenue, near Brambleton, burst from the additional pressure; that section, however, was cut off and we continued the test.

The subsequent tests for capacity against a head of 90 pounds per square inch, were made of short duration, 30 minutes each; long enough to insure the capacity of the engines to deliver against that head. The small engine was tested on the 12th inst. The mean speed of piston was 119 feet per minute, and the fluid resistance inclusive of suction was 89.207. The piston speed varied between 115 and 130 feet a minute; and the pressure in the main varied about 3 pounds each side of 89 pounds.

During the limited period of this pressure test, it was not practicable to regulate the speed or pressure closer. The steam pressure at the engine was 106 pounds per square inch, and the vacuum in the condenser was 27 inches. The speed of the engine was regulated by the cut-offs. There is sufficient reserve power in the engine and in the boilers to exceed the above quantities considerably.

The largest engine was run 30 minutes on the night of the 11th, as follows: Steam pressure of the engine, 111 pounds per square inch ; vacuum in condenser, 25.5 inches ; fluid resistance, inclusive of suction, 90.801 pounds ; speed of piston in feet per minute, 119.

* * * * * * *

FULFILLMENT OF THE CONTRACT.

The contract has been fulfilled both in the capacity and the economic performance of the plant.

With great respect,
Your servant,
(Signed,) G. W. BAIRD.

NORFOLK, VA., May 21st, 1892.

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